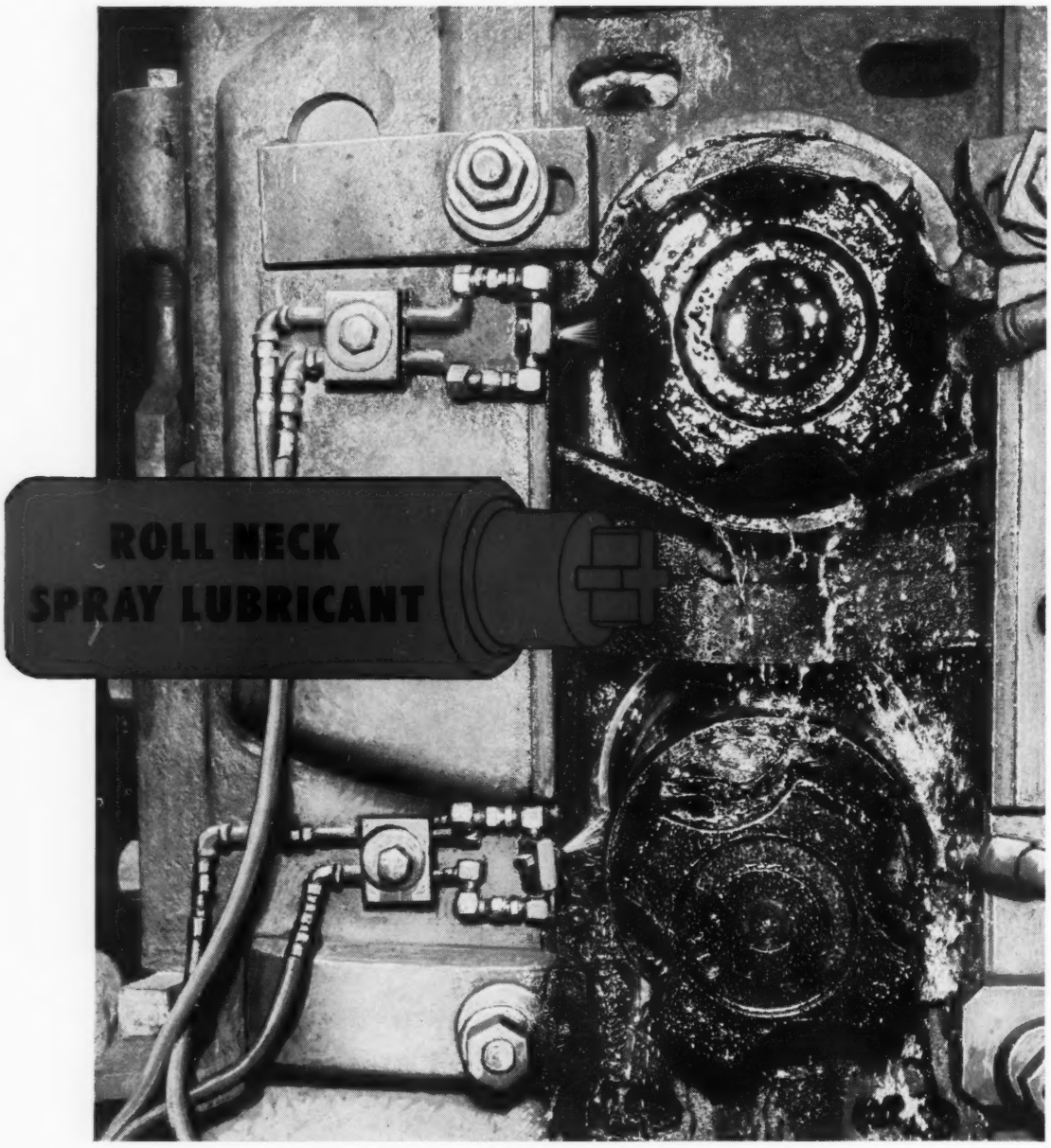
Vol. 15, No. 6 June, 1959

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

Grease Testing

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



SPRAY LUBRICATION OF ROLL NECK BEARINGS CUTS COSTS FIVE WAYS:

1. Blooming, Slabbing and Billet Mill Bearing life extended.
2. Thousands of dollars saved by reducing Mill Lining delays and downtime.
3. Shop work on bearings cut to a minimum by elimination of machining and grooving.
4. Lubricant consumption greatly reduced. Every ounce of lubricant utilized.
5. Roll Necks need no longer the constant finishing or dressing.

First adapted in 1949 and now being used by major steel companies for this important and severe application, KLINGFAST, a lead base fluid lubricant retains its film strength (4 gram Timken test, 30 minute minimum —43 lbs. L.A.L.)

For complete data, write for Bulletin K-26

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Volume 15, No. 6
June, 1959

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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Cover photo courtesy International Lubricants Corp.

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14th Annual Meeting Banquet

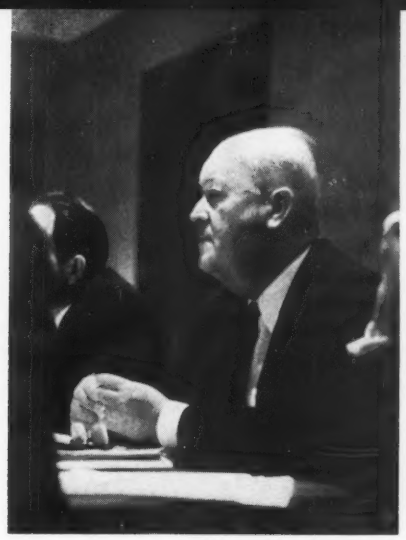
AT THE ANNUAL MEETING

(Above) Exhibitors reported much interest among members viewing the lubrication exhibits.

(Below) Registration was handled efficiently under the direction of Helen Shinovich (at right) of the National Office staff.

(Below) Colonel F. L. Gray, U. S. Steel Corp., discusses properties of lubricants at the lubrication engineering education course.

PHOTO HI-LITES FROM THE 14TH ANNUAL MEETING



Dr. A. B. Wilder, 1959-60 ASLE President, presides over first board meeting.

Buffalo, N. Y. April 21-23, 1959

(l. to r.) T. W. Havely, A. E. Cichelli, A. B. Wilder, J. D. Lykins, L. B. Sargent.

(l. to r.) L. B. Sargent, L. O. Witzenburg, D. P. Thomas, P. A. Asseff.

(l. to r.) C. L. Willey, E. G. Jackson, C. C. Blaisdell, A. B. Two, J. Boase, H. W. McCulloch, Jr.

(l. to r.) R. J. Torrens, M. M. Gurgo, B. T. Harding.

AWARDS PRESENTATIONS

M. E. Merchant (left) is presented the National Award, for his outstanding contribution in the lubrication engineering field, by J. D. Lykins.

Former President Lykins turns over president's gavel to new President A. B. Wilder.

Past President J. O. McLean awards J. D. Lykins with past-president's plaque. Lykins is now chairman of the Presidential Council.

Each year at the Annual Meeting, awards are presented to deserving members of ASLE. This year, as every year, much time and thought went into the choice of people to receive these special awards. Those considered had to have given exceptionally of themselves and their knowledge in order to further the understanding of all those in the field of lubrication. These men have contributed greatly to the Society, the profession and themselves.

John Boyd and A. A. Raimondi receive the Captain Alfred E. Hunt Memorial Award for the best ASLE paper presented during the past year.

The Walter D. Hodson award for the best paper written by an ASLE junior author is awarded to F. L. Reynolds.

A. E. Cichelli receives the Wilbur Deutsch Memorial Award for his paper "A Lubrication Engineer Surveys Steel Mill Gearing" chosen the best paper on the practical aspects of lubrication.

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1959-60 Officers Named at Confab

The 1959-60 national officers and directors of the American Society of Lubrication Engineers were announced at the 14th Annual Meeting in Buffalo, N. Y.

Dr. A. B. Wilder of the E. I. duPont Co., Chicago, Ill. succeeds J. D. Lykins, Wheeling Steel Corp., Wheeling, W. Va. as president. Lykins was named chairman of the Presidential Council, and L. O. Witzenburg, Farval Corp., Cleveland, Ohio was elected vice president at large.

Other new officers include A. E. Cichelli, Bethlehem Steel Corp., Bethlehem, Pa., secretary and C. C. Blaisdell, Penola Oil Co., Wheaton, Ill., treasurer. C. L. Willey was reappointed executive secretary.

Newly-elected regional vice presidents include D. P. Thomas, Columbia-Geneva Steel Co., Western Region, and J. Boase, Lincoln Engineering Co. Canada Ltd., Toronto, Canadian Region. Regional vice presidents retaining their office are: D. F. Wilcock, General Electric Co., Schenectady, N. Y. Eastern Region; P. A. Asseff, Lubrizol Corp., Cleveland, Ohio, Central Region, and H. W. McCulloch, Jr., Shell Oil Co., Chicago, Midwest Region.

New additions to the board of directors for a three year term are: Earl G. Jackson, General Electric Co., West Lynn, Mass., and Howard T. Peeples, Timken Roller Bearing Co., Canton, Ohio. L. B. Sargent, Aluminum Co. of America, New Kensington, Pa. was elected industrial director for a two year term.

Directors continuing their term of office include: R. J. Torrens, Eastman Kodak Co., Rochester, N. Y.; E. E. Bisson, NASA, Cleveland, Ohio; B. T. Harding, Midwest Oil Co., Minneapolis, Minn.; A. B. Two, Motor Oils Refining Co., Chicago, and M. M. Gurgo, Standard Oil of New Jersey, Pittsburgh, Pa.

Annual Meeting Keynote—"Change"

ASLE's Board of Directors gave attention to a far-flung agenda at the Society's 14th Annual Meeting.

Election of officers, expansion of editorial coverage of *Lubrication Engineering*, constitutional changes, and an increase in membership dues occupied the attention of the officials during three days of meetings at Buffalo, N. Y.'s Statler-Hilton Hotel.

Among those elected to office were L. O. Witzenburg, vice president at large, A. E. Cichelli, secretary, and C. C. Blaisdell, treasurer. All new officers and directors were invited to attend the Board meeting.

Recognizing the need for greater dissemination of lubrication information, the Board authorized early hiring of a engineering writer for *Lubrication Engineering*. In addition, articles for the magazine are being solicited. Presentation of papers at a meeting is no longer a prerequisite for publication in either *Lubrication Engineering* or *ASLE Transactions*. (see Editorial on page 242).

ASLE members will soon be asked to vote on the constitutional revisions made by the Board under the advisement of the By-Laws committee. A set of by-laws and operating rules is being developed for future Board consideration.

Dues increases effective January 1, 1960 will be: Membership dues $20 (formerly $15), Associate Membership dues $20 (formerly $7.50), and Sectional Sustaining Membership dues $35 (formerly $25).

Society's Offices Will Be Moved To New Location

The National Headquarters of ASLE is being moved to a new location. The newly decorated, air-conditioned offices are located at 5 N. Wabash Ave. on the 17th floor.

Calvert L. Willey, ASLE executive secretary, signs lease for Society's new offices.

The building in which the new ASLE offices are located is found in the hub of Chicago's loop.

This new location (still in the loop area) is readily accessible to the business area of Chicago. Complete operations, all on one floor, will permit greater efficiency in handling the volume of mail that is processed in the National Office.

While the overall space is the same as that presently used, the unencumbered area will make for smoother operation. The new office location, adjacent to other engineering societies and related groups, in a dignified area, lends considerably to the Society's prestige. The staff members will move to the Wabash Avenue location during the latter part of June. ASLE members visiting the Chicago area are invited to drop in and see the new headquarters.

Job Opportunities For Engineers On Upswing

The Department of Labor's latest report shows professional engineering job vacancies filed with local employment offices more than doubled between February, 1958 and February, 1959,—2,146 to 4,418. The previous month, January, 1959, 4,312 openings for professional engineers were available. The heaviest demand, by branch, was electrical, followed by mechanical and aeronautical.

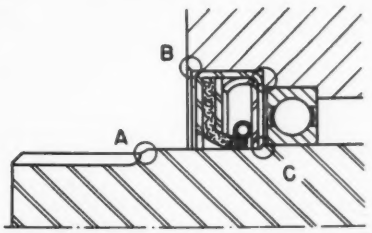
NATIONAL OIL SEAL® LOGBOOK

4 common shaft sealing conditions

. . . and engineering tips that can help you "design-in" better oil seal performance

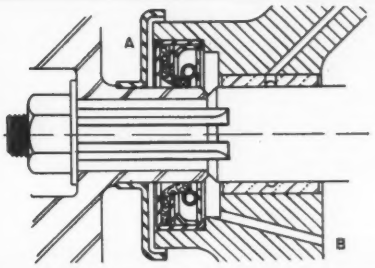
CONVENTIONAL INSTALLATION

Here a standard-design single lip seal retains lubricant and excludes normal dirt, dust and moisture. Sealing lip points in since seal's principal job is retaining oil or grease around bearing. Note that shaft is stepped and chamfered at "A" to prevent damage to sealing lip during installation. At "B", bore is chamfered to facilitate seal entry. At "C", counterboring insures accurate positioning of the seal.



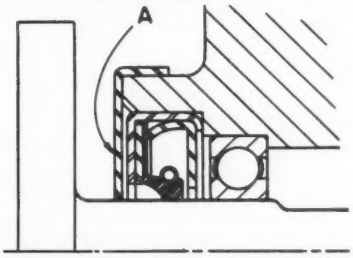
HEAVY DIRT CONDITIONS

Here is a commonly used method of protecting the seal and increasing seal life on applications subjected to extreme dirt conditions. The guard baffle at "A" is welded or swaged to the wheel hub to exclude the major portion of dirt and dust. The drain hole at "B" relieves pressure at the sealing point. In addition to the guard baffle, many manufacturers employ a dual-lip seal to insure bearing protection under extreme dirt conditions.



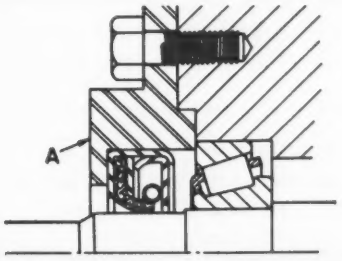
SEALING LONG, HEAVY SHAFT

Many cases of so-called "seal failure" are due solely to poor installation techniques. While today's seals are rugged, they can be rendered non-serviceable if distorted out of round, cocked in the bore, or if the sealing lip is torn. To protect the seal against such physical damage during installation involving a long shaft, a seal protector as shown at "A" may be mounted on the hub O.D.



INSUFFICIENT DEPTH TO MOUNT SEAL

Where the housing does not provide sufficient depth for counterboring, or where seal installation would be difficult or likely to damage the seal, a separate mounting member ("A") can be employed. As before, the shaft should be chamfered to prevent damage to the sealing lip during installation.



NATIONAL SEAL
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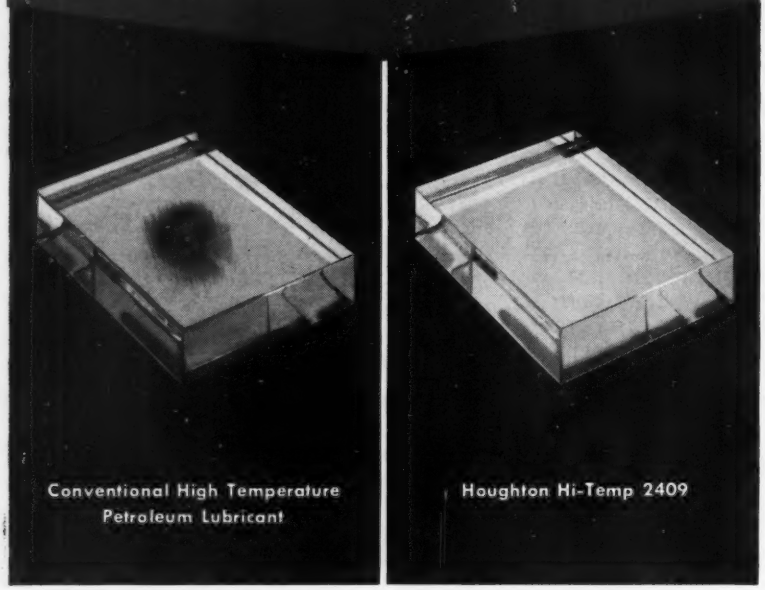
Scratch Test Proves NEW Hi-Temp 2409 Lubricates at 1000°F.



These two pieces of plastic were each rubbed with the residue of grease after it had been exposed to a temperature of 1000°F.!

This scratch test tells the story! The piece rubbed with *the lubricating residue of Hi-Temp 2409* is as clear as crystal! The piece rubbed with the hard, gritty residue of conventional high temperature grease is badly scratched.

This proves what happens to bearings that operate at temperatures from 600°F. to over 1000°F. With new Houghton Hi-Temp 2409 grease, bearing wear and breakdown due to inefficient lubrication is eliminated!

If you have a high temperature lubrication problem, it will pay you to investigate this new grease—Hi-Temp 2409. Call your Houghton Man for a demonstration or write E. F. Houghton & Co., 303 W. Lehigh Ave., Philadelphia 33, Pa., for your copy of Houghton's new booklet on "Thermal Lubrication".

HIGH TEMPERATURE LUBRICANTS

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PERSONALS

Bray Oil Co. has announced the appointment of **Dr. M. Z. Fainman** as director of research. Formerly executive engineer with Inland Testing Laboratories, Dr. Fainman is nationally known for his application of the principles of physical organic chemistry to the synthesis of lubricant esters and hydrocarbons.

Dr. Henry E. Mahncke has been appointed chemist for SKF Industries, Inc. He will be in charge of the ball and roller bearing manufacturer's chemical research section of the laboratory.

Robert T. Daily has been appointed manager of marketing for the Silicone Products Dept., General Electric Co.

John R. Sabina has been named manager of the tetraethyl lead and planning section of the Du Pont Co.'s petroleum chemicals division.

Jack Bradley has been appointed as an assistant sales manager of the industrial division of Lincoln Engineering Co. His specific responsibilities will be sales of industrial lubricant pumps and materials dispensing equipment manufactured by Lincoln.

Paul J. Reeves is now vice president in charge of sales for the Timken Roller Bearing Co. Also, four executive changes in the sales division include Robert G. Wingerter, director of sales; S. T. Salvage, general manager of the automotive division; N. H. Peterson, advertising manager, and W. R. Covey, assistant advertising manager.

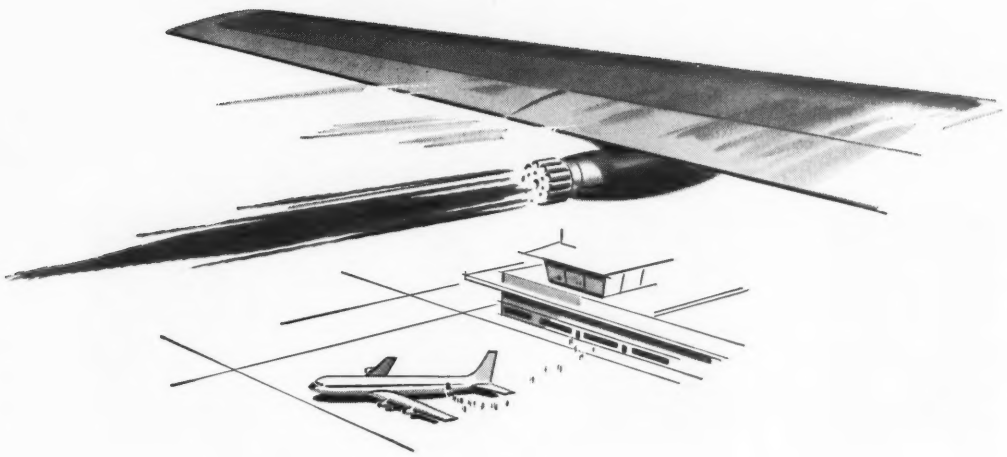
Joseph Dixon Crucible Co. has elected H. Edward Ehlers and Warren A. Zimmer as senior vice presidents and Sherwood B. Seeley as technical director.

Jean Groff, head of the lubricants department of the Institute Français du Pétrole, died February 23 in his native France.

INDUSTRY NOTES

International Lubricant Corporation has moved its executive offices and laboratory facilities into a two-story building just northwest of the city limits of New Orleans. The move was made primarily to bring together all of ILC's extensive laboratory and testing equipment, much of which had been scattered throughout the plant on the same property since the destruction of the former lab by fire on December 13, 1955.

In 1946, ILC was purchased by the Shell Oil Co. William H. Saunders and Harold M. Fraser continued as the operating officers and, last year, Benjamin G. Symon (former industrial director of ASLE), who had been manager of Shell's Industrial Products Dept., was added to the management team as vice president of sales.



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Address _____
City _____ *State* _____

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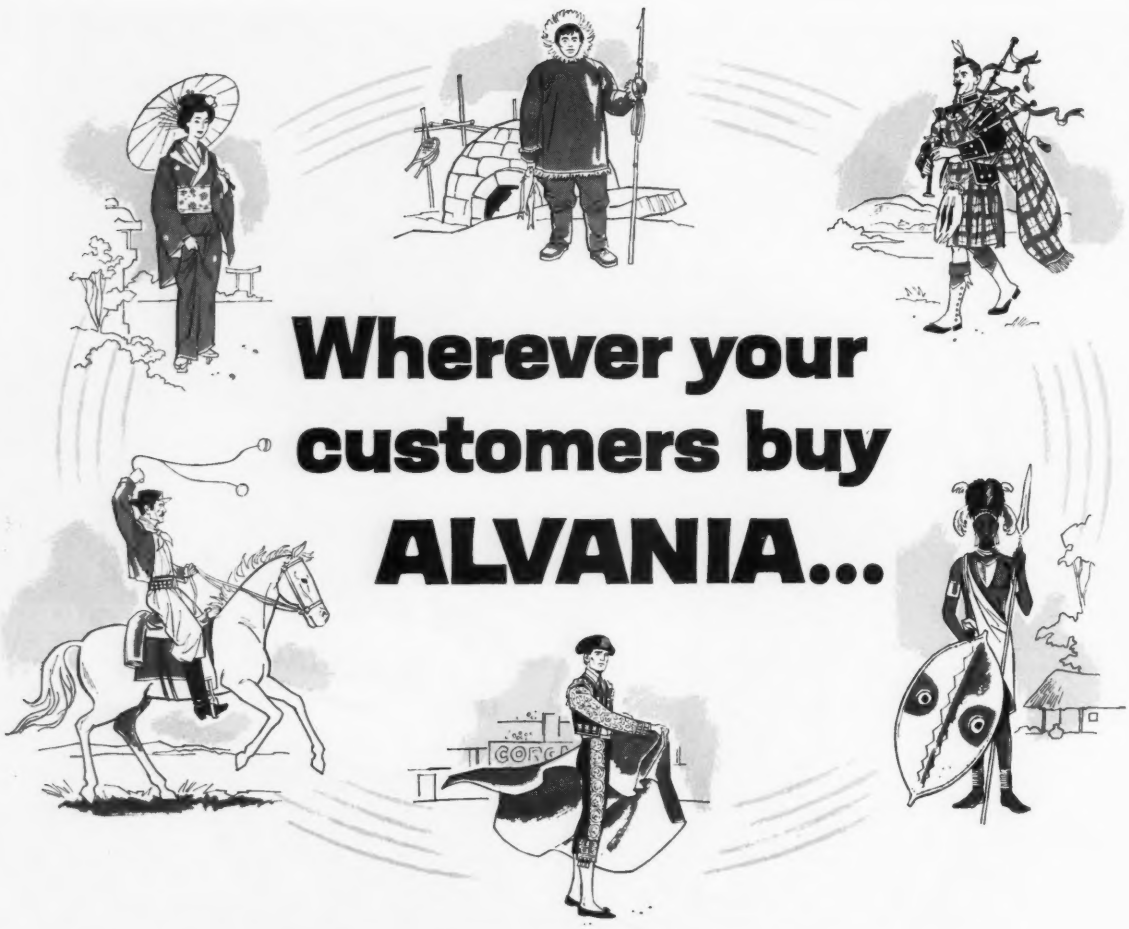

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Shell Voluta Oils—for high-speed quenching with maximum stability

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For complete information on this truly multi-purpose grease, write Shell Oil Company, 50 West 50th Street, New York 20, New York, or 100 Bush Street, San Francisco, 6, California. In Canada: Shell Oil Company of Canada, Limited, 505 University Avenue, Toronto 2, Ontario.

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the truly multi-purpose lubricant

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AT LESS COST WITH

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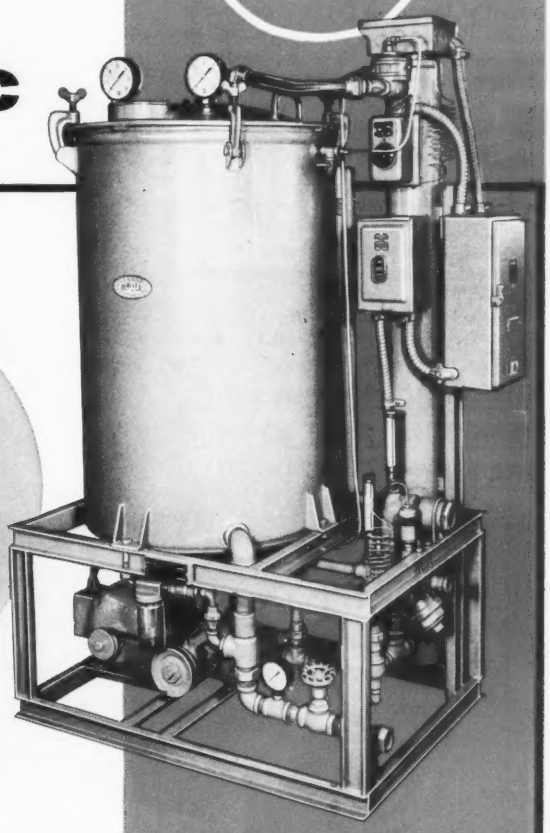
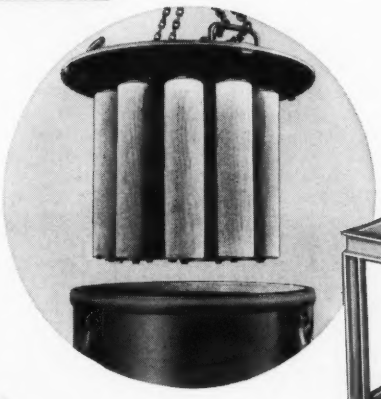
Removes water AND dirt. More efficient than centrifuging. Does not remove additives or inhibitors. Breaks emulsion as it "drycleans." Saves man hours, requiring only a part-time operator. Mail coupon for 12-page FIL-THERMO-VAC FACT BOOK.





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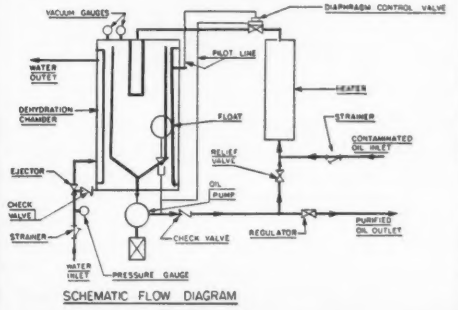


SCHEMATIC FLOW DIAGRAM

BOWSER, Inc. DATE_____
FORT WAYNE, INDIANA

Gentlemen: Send FACTS ABOUT FIL-THERMO-VAC to:

Name_____

Company_____

Address_____

City_____ State_____

NEW PRODUCTS

NEW COOLANT UNIT INCREASES PRODUCTION

A compact, low cost coolant unit especially designed for small precision drilling, tapping, milling, sawing, grinding, etc., operations is now available. Type J "Trico-Mist" is equipped with a two ft. Neoprene coolant line and mist generating nozzle. A mounting stud is provided for easy installation. It has dual valves, one to control air and the other to control flow of liquid coolant dispensed at the tip of the mist generating nozzle. Because mist is achieved by an atomizing principle, there reportedly is no air pressure in the reinforced shatterproof 8 oz capacity plastic reservoir at any time. Coolant supply is always visible. Since small jobs only require a small fine mist, one filling will last eight hours and longer when used constantly. Each unit includes a magnetic nozzle holder having an adjustable arm so that the coolant can be directed to the cutting edge of the tool. The tip of the nozzle is protected against damage by a special guard. All metal parts are solid brass and brush finished. Details are available from Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wis.

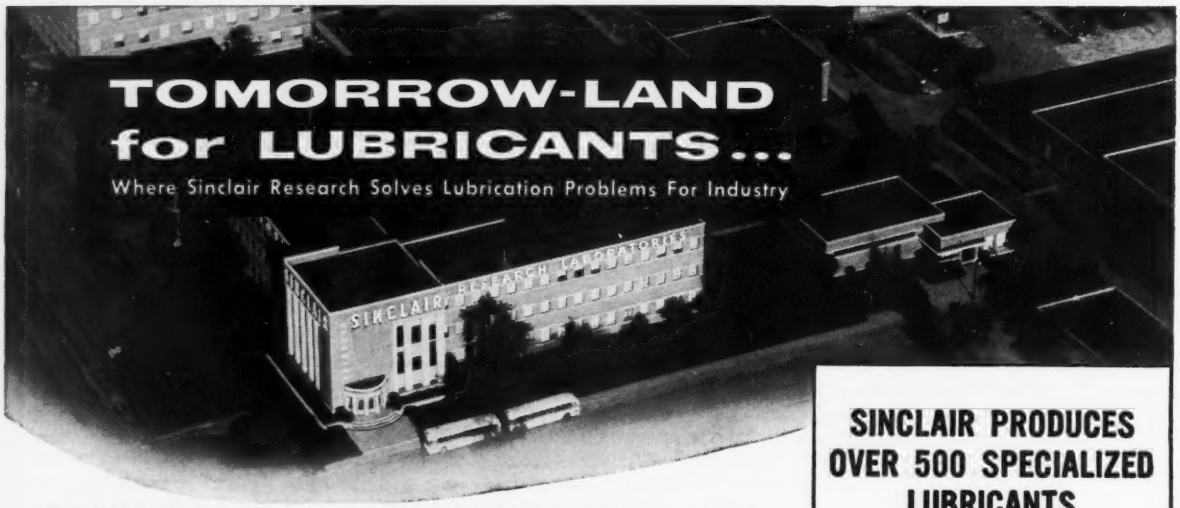
LEAK AND CLOG-FREE VALVES DEVELOPED

A new series of screw stem shut-off valves, designed to prevent both leakage to atmosphere, and clogging of activating threads by viscous fluids, has been developed. According to the manufacturer, the new 30V-50V series features a single Buna "N" O-ring which by its designed position eliminates leakage problems and maintenance costs and provides improved performance over a longer period of time. The new unit is available in standard models for use with fuel oil, gasoline, kerosene, air, water, mild alkalies, acids, alcohols and coolants up to 500 psi at —40F to +250F in $\frac{1}{8}$ in and $\frac{1}{4}$ in IPT sizes. Additional information is available from Anderson Brass Co., 102 S. Campbell Ave., Detroit, Mich.

HYDRAULIC FLUID RESISTS FIRE

A new fire-resistant hydraulic fluid, Shell 3XF mine fluid, for use in mining equipment has been introduced by Shell Oil Company.

Unlike mineral oil now commonly used in mine operations, the new fluid emulsion has a built-in fire extinguisher —water. Each microscopic droplet of water is surrounded by a tough film of lubricating oil. On exposure to open flame, high voltage arcs or hot metal, the fluid's water content is immediately released as a steam blanket to put out any resulting fire. It reportedly meets all requirements for a high quality hydraulic fluid. The U. S. Bureau of Mines has evaluated and approved the product for use in mining machinery.



Located at Harvey, Illinois, is one of the most extensive installations of its kind in the world—Sinclair Research Laboratories. These facilities are an important part of Sinclair's investment in the future. Here is where Sinclair engineers and chemists work to develop new products and improve the quality of existing ones. At these famous laboratories were developed the Sinclair lubricants now solving difficult problems in all branches of industry. If you have a special lubrication problem, write today to Sinclair Refining Company, Technical Service Division, 600 Fifth Avenue, New York 20, N. Y.

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PRODUCT LITERATURE

NEW FOLDERS DESCRIBE PUMPS

Four new 4-page brochures and two 2-page units describe the complete line of internal gear rotary pumps offered by the Wayne Pump Co., Division of Symington Wayne Corp., Fort Wayne, Ind. Inside pages contain a descriptive paragraph, construction options and details, specifications for each model in the series, ordering instructions, assembly recommendations depending on applications, and dimensional details of the pump. The back page provides dimensions for V-belt and gear driven assemblies. Sets of this literature may be obtained by request.

THERMOPLASTIC SELECTION

"How To Select A Thermoplastic" is the title of the featured article in the current issue of *Plastics Fabricator*, external house organ of the American Agile Corporation. The special article points out that even within the family of thermoplastics, certain types are best for given applications, and therefore, user companies must understand which types will give them the best service under expected operating conditions. Copies may be obtained from the American Agile Corp., P. O. Box 168, Bedford, O.

BROCHURE EXPLAINS CENTRALIZED LUBRICATION

Application of modern centralized lubrication to reduce industrial operating costs and improve production is discussed in "A Report to Management." The fully-illustrated 16-page booklet explains the role of centralized lubrication —its advantages over manual methods of greasing and oiling industrial machinery; three economic areas of plant management where it can effect substantial savings; five practical steps for converting to automatic lubrication; case histories of uses in all major industries, and detailed application data on basic centralized systems. Copies may be obtained from Lincoln Engineering Co., Industrial Div., 4010 Goodfellow Blvd., St. Louis 20, Mo.

COOLANT SYSTEMS

A new bulletin on Dravo-DeLaval lubrication and coolant systems for the steel, non-ferrous metal and paper industries has been produced by Dravo Corporation, Pittsburgh, Pa. Featuring illustrations of recent installations in the various industries served, the booklet also lists component parts supplied by the company as well as convenient specification tables on oil reservoirs, pressure tanks, and the company's Unilube prefabricated lubrication systems. Copies may be obtained by writing for Bulletin 1451.

RUST PREVENTION BOOKLET

A 20-page booklet prepared by Rust-Lick, Inc. presents basic information on rust, its causes, and application of preventives. Included is factual data on the various types of the manufacturer's rust preventing liquids, their recommended applications, and a graphic reproduction of the product's rust preventing qualities. The booklet is available from Rust-Lick, Inc., 755 Boylston St., Boston 16, Mass.

OIL SELECTION BULLETIN

A new bulletin, "Lubrication of Roller and Silent Chain Drives," covers principles of lubrication, maintenance, and oil selection. Oil selection charts are included. Copies are available from Sun Oil Company, Industrial Products Dept., 1608 Walnut St., Philadelphia 3, Pa.

CATALOG OF SWITCHES

Newly revised Catalog 90b describes the standard line of Honeywell mercury switches, manufactured by Micro Switch, a Division of Minneapolis-Honeywell Regulator Co., Freeport, Ill. Included are photographs, dimensions, electrical ratings, drawings, application information and complete technical data. Honeywell mercury switches provide dependable, quiet, trouble-free operation in a wide variety of uses—from simple door interlock mechanisms to intricate space craft guidance systems.

SILICONE RUBBER GUIDE

A silicone rubber selector chart, CDS-145, is now available from the Silicone Products Dept., General Electric Co., Waterford, N. Y. Reportedly one of the most complete silicone rubber specification guides available today, this publication is designed to assist designers and engineers in selecting the proper type of silicone rubber for their particular requirements. The publication contains comprehensive data on applications, typical properties, primary classes and standard industry and military specifications.

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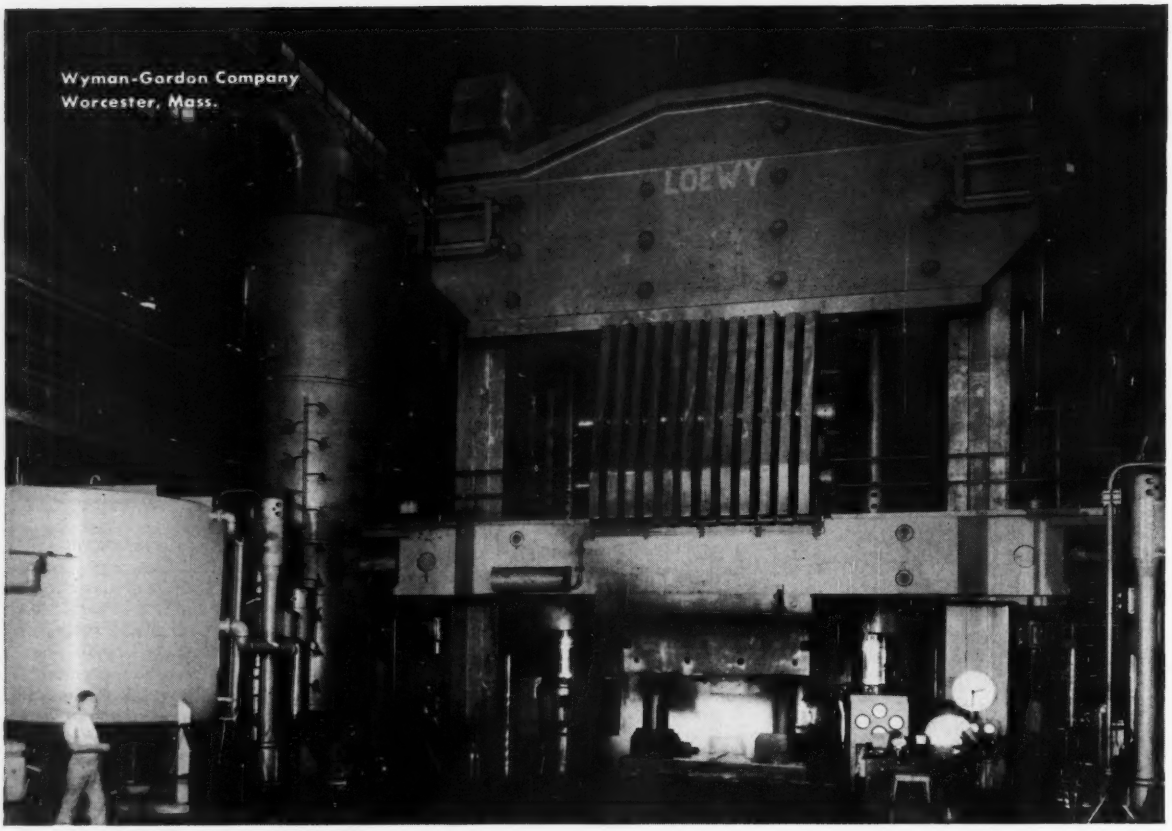

Operation Big Squeeze:

World's largest presses lubricated by Cities Service Pacemaker Hydraulic Oils

Once a forging operation for horse-drawn carriages, Wyman-Gordon Company now operates the world's largest presses, with the newest one capable of exerting 50,000 tons pressure.

Reason for this tremendous press is the increasing need for strong, lightweight metal parts for aircraft. Today, as for many years, no military or transport aircraft takes to the air without Wyman-Gordon forgings.

With its huge, hydraulically operated presses, Wyman-Gordon is able to forge huge pieces with one big squeeze. Sometimes a whole unit for a landing-gear strut. Or one mammoth wing spar.

The press that exerts 50,000 tons pressure is described by Wyman-Gordon as "the largest single machine ever conceived"—and few people would argue the point after seeing it.

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It is hardly necessary to point out the incredible demands that such machinery makes of its hydraulic lubricants... and certainly it is self-evident that if Cities Service Pacemaker Oils can perform satisfactorily in the world's largest presses, they'll do the same for you. Talk with a nearby Cities Service Lubrication Engineer. Or write: Cities Service Oil Co., Sixty Wall Tower, New York 5, N. Y.

Entire Rig Is Operated by One Man with the few simple controls shown here. With a flick of his fingers he can put on a squeeze of 50,000 tons. Result is extremely strong lightweight forgings for aircraft.

Lubrication Foreman has mammoth responsibility. Here, he checks dispenser for Cities Service EP 20 Lubricant... used on main bed and columns of large presses and other areas subject to extreme pressures.

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Vol. 15, No. 6

Lubrication Engineering

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One of the most important ways of promoting the advancement of knowledge is through the papers which appear in the various Society publications. To carry out this purpose as effectively as possible, ASLE is constantly looking for new and useful papers on every aspect of lubrication including design, research, application, and maintenance. Authors are encouraged, therefore, to submit three copies of their papers to the Society whenever they feel that they have material which would be of interest to the readers of either *Lubrication Engineering* or *ASLE Transactions.* All such papers should be sent to Society headquarters. This material will be considered for publication, and the authors will be advised as to whether or not their papers will appear in print. If an author so desires, he may request that a paper be considered for presentation at one of the Society's meetings in addition to publication. Such a presentation, however, is not a prerequisite for publication.

In order to assist authors in obtaining as rapid publication as possible, an Author's Guide is available from Society headquarters upon request.

John Boyd,
Editor

Designing and Testing Journal Box Seals

W. L. Stewart,

American Brake Shoe Company, Mahwah, New Jersey

In recent years there has been an increase in demand for a journal box seal which will satisfactorily seal dirt, dust and water from entering the journal box and prevent the loss of oil.

The current standard dust guard consists of a 5/8 inch piece of plywood, or other material, cut to a standard shape and bored to fit the axle. The seal fit opening in the standard wooden dust guard is 1/16 inch larger in diameter than the seal fit section of the axle. This type of dust guard does not seal oil within the journal box and does not prevent water, dust or small stones from entering the journal box. The leaky and ill-fitting conditions of the present standard dust guard are contributory factors leading to contamination of the journal box. This contamination causes rapid bearing wear, cut journals, water logging of lubrication medium and oil leakage. The increased use of lubricating pads[1] has shown a noticeable increase in oil consumption. This increase in oil consumption is caused by the loss of oil at both the rear and front of the journal box. Waste lubrication provided a dam at the rear of the box and acted as a baffle in the journal box cavity which restricted the loss of oil. Although the back roll of waste acted as a dam at the bottom of the dust guard cavity, it also caught the surplus oil from the bearing. This surplus oil dripped from the rear of the bearing and was channeled by the back roll of waste to the seal fit section of the axle. A large portion of the oil lost from waste lubricated journal boxes was lost in this manner. The lubricating devices supplanting waste allow oil to splash back and forth in the journal box. Railroads have expressed a concern over increased oil loss through the ineffectively sealed dust guard opening. An efficient journal box seal would result in savings in oil consumption.

The problems encountered in attempting to seal the back end of the railway journal box are different from those encountered in any other industrial sealing application. A journal box seal must be capable of sealing two areas at the rear of the journal box. First, the oil must be prevented from leaking out of the box on the seal fit section of the axle, and second, loss of oil must be prevented through the dust guard well. The front of the box should be sealed to prevent loss of oil out of journal boxes of cars that are turned over in a car dumper and to prevent the loss of oil flung off by the axle collar.

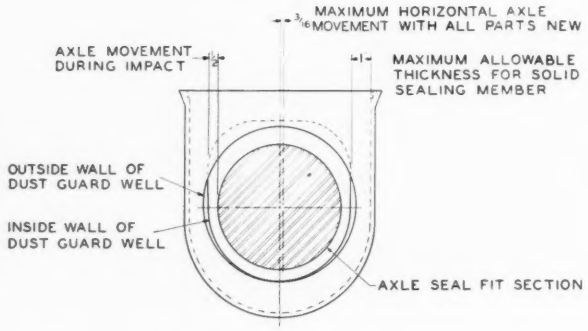
MAXIMUM HORIZONTAL AXLE MOVEMENT WITH ALL PARTS NEW
3/16
AXLE MOVEMENT DURING IMPACT
1/2
1
MAXIMUM ALLOWABLE THICKNESS FOR SOLID SEALING MEMBER
OUTSIDE WALL OF DUST GUARD WELL
INSIDE WALL OF DUST GUARD WELL
AXLE SEAL FIT SECTION

Fig. 1. Rear view dust guard cavity.

The railroad journal box presents a particularly different seal application. The rotating axle has a lateral movement in and out of the journal box. In addition, the axle moves in both horizontal and vertical planes. The movement of the axle in the horizontal and vertical plane is caused by the forces developed during acceleration, deceleration, braking with single brakes and impact during switching operations. The normal horizontal movement may be as great as 3/8 inch when the parts of the journal box assembly are new. During impact the bearing and wedge are often displaced, and the axle moves horizontally until it contacts the inner wall of the dust guard well (Fig. 1). The movement of the axle in the horizontal plane is not necessarily uniform, and often an angulation takes place. The angulation may be caused by the normal tracking of the truck when negotiating curves or by variations in wheel diameters. This axle movement presents a prob-

[1] The American Association of Railroads has recently ruled that lubricating pads will replace cotton waste as the lubricating medium used on railroad car journal boxes. A lubricating pad consists of an oil feeding medium to feed oil to the journal and a resilient member to maintain positive contact.

Presented before the 13th Annual Meeting of the ASLE, April 1958, at Cleveland, Ohio.

lem as the sealing member is held by the dust guard well which does not move integrally with the axle. Fig. 2 shows a standard 5½ x 10 AAR Journal Box assembly with all parts new. By comparing the two portions of Fig. 2 which show the maximum lateral movement which may be expected, it can be seen that the journal is free to move laterally one inch. Fig. 3 shows a standard AAR journal box with all parts at condemning limits. Comparison of the top and bottom sections of the figure shows that the axle is free to move laterally 2⅜ inches.

An inspection of journal boxes in the field has shown that excessive lateral wear will cause the wheel hub to contact the rear of the journal box and grind away a portion of it. It is extremely difficult to design a seal which will contact the seal fit of the journal at all times under these conditions. If a compromise is made and the seal is designed to be effective for the maximum number of applications, the seal must be capable of riding on and off the seal fit without damage to either part. Obviously, the seal fit of the axle cannot be 100 per cent effectively sealed in this manner. The standard dust guard does not operate satisfactorily under these conditions. The standard dust guard fits loosely in the dust guard well and drops to the bottom of the well when it rides off the seal fit. On the return lateral stroke of the axle the dust guard is struck by the sharp edge of the seal fit section of the axle and damaged.

Another design factor which must be considered is the impact involved during switching and humping operations. In a new 5½ x 10 journal box the established tolerance is such that the maximum width of a solid sealing member is limited to one inch (Fig. 1). Wear of the dust guard section of the journal box in service further reduces the maximum width to which the solid sealing member is limited. If a solid member of the sealing device is not smaller than the aforementioned dimension, the member would be crushed during impact.

Another design factor which must be considered is the vertical movement of the seal in the dust guard cavity. When bearings are installed it is standard practice to jack the journal box until the seal fit section of the axle contacts the bottom of the dust guard well. The seal must be able to shift into this cavity without damage. Fig. 4 shows the maximum allowable wear which may occur in the vertical direction. The centerline of the axle can vary 11/16 inch.

The opening in the dust guard cavity in which the seal must operate is nominally ¾ inch. The tolerance allowed is ±1/16. In practice it has been found that some of the boxes actually vary in greater amounts than this. In order to effectively seal this portion of the journal box the seal must be designed to function satisfactorily through this range of thickness.

A seal made of rigid, smooth surface material obviously can not seal against the cast finish walls of the dust guard well. The seal must be of a material which is resilient enough to conform to the variations cast in the cavity. If the material is of resilient nature it must not take a permanent set when installed or the effectiveness of sealing around the dust guard well opening will be lost.

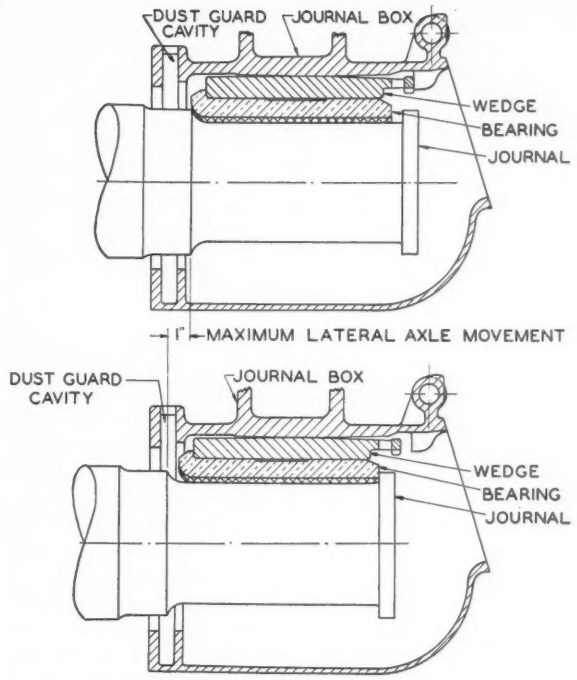

Fig. 2. Maximum lateral axle movement with all parts new.

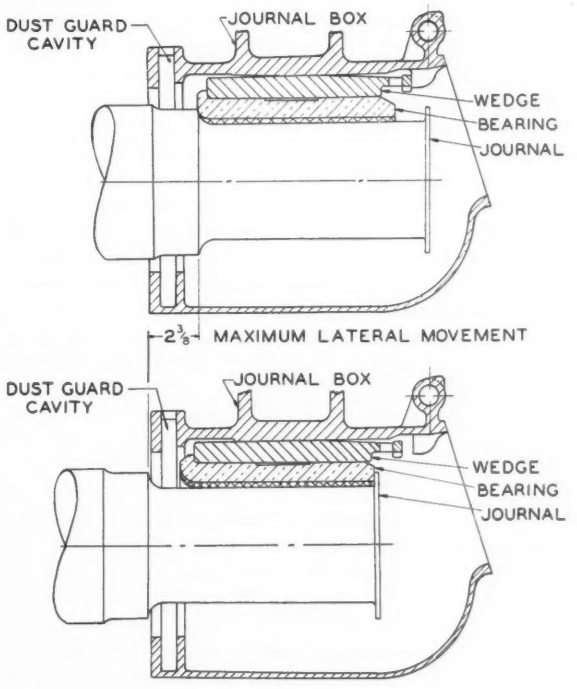

Fig. 3. Maximum lateral movement with all parts at condemning limits.

Some other factors which must be considered are the condition of the seal fit surface of the axle and its dimension. The seal fit section of the axle is frequently damaged during impact when it strikes the side walls of the dust guard well. When the journal is turned

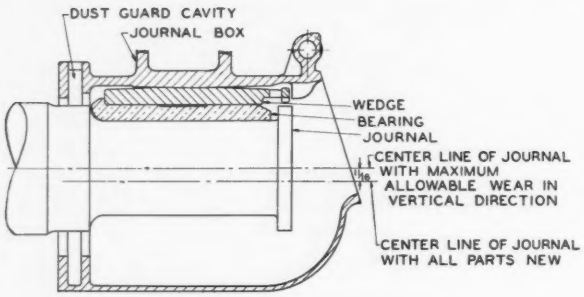

Fig. 4. Maximum allowable wear in vertical direction.

down, the seal fit is usually reduced in size (both in diameter and length). An inspection of railroad axles in the field has shown that some seal fit sections are reduced as much as ⅜ inch in diameter. Designing a seal which will run on a badly gouged surface which may vary ⅜ inch in diameter is an extremely difficult problem.

The last design features which must be considered are those of material. Material used for the journal box seal must be oil and water resistant. The AAR Specification M-903-47 for dust guards calls for immersion of the dust guard in car oil and water at a temperature of 160F to 175F for not less than 140 hours without showing signs of separation. The seal must operate in a temperature range which may vary from —40F to 225F. For short periods of time the seal must be capable of operating in the vicinity of 300F. The materials used must be capable of withstanding these temperature variations without being adversely affected. The portion of the seal which contacts the seal fit of the journal must have good wear resistance and a low coefficient of friction. Wear resistance is an important property in sealing the journal box since the seal fit of the journal is often rough and the seal is continuously subjected to the dust and dirt thrown up from the rail bed. If a resilient material is used to seal the sides of the dust guard well, this material must be able to withstand a compression set so that separation will not take place between the walls of the well and the sealing member.

TESTING

DESCRIPTION OF TEST EQUIPMENT

The testing devices which are described are equipped for testing seals designed to operate in a standard 5½ x 10 AAR journal box. Fig. 5 shows the oil sealing test assembly. The test stand is equipped with lateral motion which is cycled at the rate of 50 cycles per minute, and has a travel distance of ¾ inch. The journal is lubricated with a single wheel free oiler[1] placed at the center of the journal but the test stand does not include a bearing and wedge. The single wheel free oiling device is used for two reasons—first, to eliminate the bearing wedge and lubricating pad friction, and second, to furnish an abundant supply of oil. The diameter of the seal fit section of the axle is 6.625 inches ±.002 inches. The journal is operated a a speed corresponding to 42 mph for an actual freight car in service, which is equivalent to a seal fit surface speed of 750 feet per minute. Finish of the seal fit section of the test axle is 15 micro inches. The stand is driven by a ¾ hp three phase motor with a simple pulley speed reduction. The lateral is accomplished by the use of a gear reduction motor and a variable speed drive.

Fig. 6 shows a special torque measuring device which is used to determine dry and lubricated running torques. This device consists of a standard 5½ x 10 journal around which is mounted a seal holding device. The seal holding device is supported in the stand by roller bearings. The cavity of the seal holding device is a standard 5½ x 10 dust guard well. The diameter of the seal fit section of the axle is 6.625 ±.002 inches. The axle is driven by a ½ hp motor with force being measured by two pull scales. The finish of the seal section of the testing device is 15 micro in. Friction horsepower developed by the seal is calculated from the formula:

$$hp = \frac{2\pi NR(F_1 - F_2)}{33{,}000 \times 12}$$

N = Revolutions per minute
R = Radius of seal holding device in inches
$(F_1 - F_2)$ = Difference of spring scale readings in pounds

TEST PROCEDURE

The standard test procedure used consists of six tests. The first three tests were run on the oil sealing test assembly.

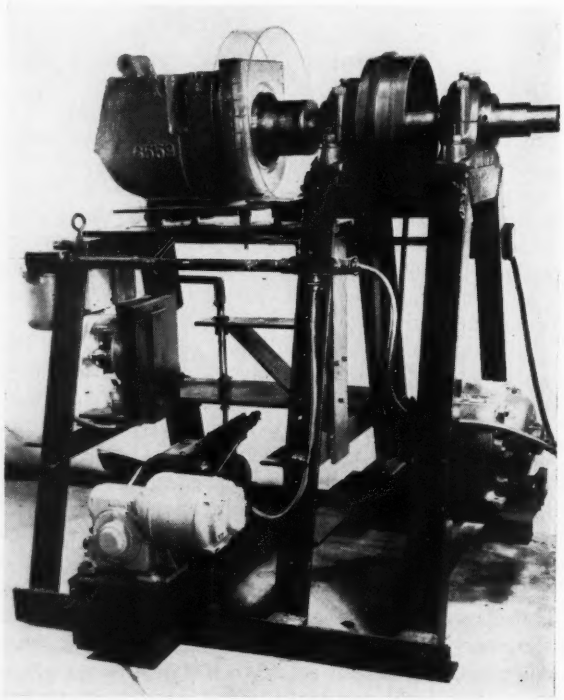

Fig. 5. Test stand with oil sealing test assembly.

[1] A single wheel free oiler consists of a round disc mounted in the journal box so that the disc is driven by the axle. The disc is partially submerged in the oil. When the journal is rotated it drives the disc thereby supplying oil to the journal.

The primary purpose of the first test is to determine if the seal prevents loss of oil through the bottom of the dust guard well. This test is run for a duration of six hours with the seal riding on the seal fit. The lateral motion is at 50 cycles per minute, test stand speed is 42 mph and the oil level at the start of the test is 2 in. This is equivalent to 64 oz of oil in the test journal box. The test stand position is level. The amount of oil which is lost is caught in a pan and measured at the conclusion of the test. In addition a rotating thermocouple equipped with a slip ring assembly pickup is placed near the surface of the seal fit and the temperature is continuously recorded during the test on a 12-point recorder. Ambient temperature is also recorded.

The purpose of the second test is to determine if the seal prevents loss of oil through the seal fit. This test is of three hours duration with the seal riding on the seal fit. The lateral of the test stand is 50 cycles per minute and the speed 42 mph. An oil level of one inch is established, which is equal to 23 oz in our standard journal box. The test stand position is at an 8 degree backward tilt. These conditions provide an excess amount of oil on the seal fit section of the journal. The oil loss is again measured and during the test a splash device is attached at the rear of the journal box to catch all oil thrown from the seal fit of the journal. The temperature of the seal seat section and the ambient temperature are recorded.

The purpose of the third test is to determine if the seal will be damaged when riding on and off the seal fit of the axle. This test is of three hours duration with the seal riding on and off the seal fit of the axle. The lateral is at 50 cycles per minute, the test stand speed at 42 mph, oil level is one inch and the test stand position is level. During the test the following recordings are made: Volume of oil lost, temperature of the seal fit, ambient temperature, condition of seal at conclusion of test.

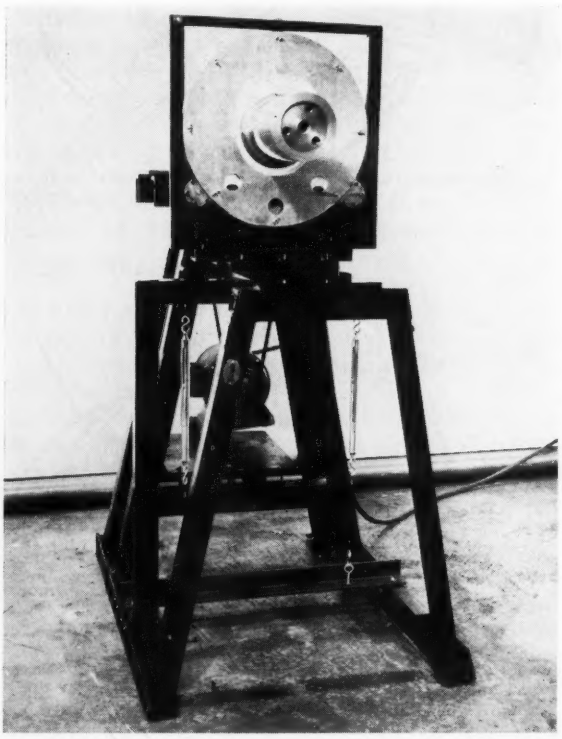

Fig. 6. Torque measuring device.

The fourth test is run on the torque testing device. The seal is placed in the torque measuring device and the journal is rotated at approximately 42 mph. The scale readings are recorded and the speed of the journal is measured. The seal is tested in this manner with and without lubrication. The horsepower drawn by the seal is then calculated.

The fifth test is to determine the effect of soaking the seal in hot oil at 175F for 140 hours, and detrimental effects are recorded.

The sixth test is similar with the exception that the seal is soaked in water at 175F for 140 hours.

For tests number one and two, a control was established by running the test stand through the standard procedure without a seal installed and with only the lateral motion operating. This test was conducted to determine the maximum amount of oil which could be lost through the bottom of the dust guard cavity under the test conditions. During test number one it was found that an average of 19 oz of oil would be lost. During test number two an average of 1½ oz was lost. Test results of the standard plywood dust guard showed an average of 12.6 oz of oil lost during test number one. During test number two an average of 11.1 oz of oil was lost and during test number three no oil was lost. In addition, during these tests the operating temperature at the seal fit at the conclusion of the test was 86F for test number one 88F for test number two and 84F for test number three. The low operating temperature of the standard wooden dust guard is due to its loose fit. Since the standard wooden dust guard is 1/16 inch larger in diameter than the seal fit section of the journal only a small portion contacts the seal fit during operation.

A second control test was run to determine the ratio of the oil loss through the seal fit section to the oil loss through the dust guard cavity of a standard wooden dust guard. A standard wooden dust guard was modified so that no oil loss would occur through the dust guard cavity. Test number one resulted in only a trace of oil lost. Test number two resulted in an average loss of 3.4 oz of oil. The percentage of oil loss through the seal fit section equaled zero per cent in test number one and 30.6 per cent in test number two.

CONCLUSIONS

The need for a good journal box seal is greater today than it has ever been. The lubricating pads have proven their usefulness by the record they have compiled in the last few years. The next problem is to find a way to satisfactorily seal the journal box and the test procedure outlined in the paper has been set up to assist in solving the problems.

From the design considerations mentioned and from tests made thus far, it can be seen that a 100 per cent effective seal for the AAR Journal box is prac-

(*Cont. on p. 264*)

Investigation of the Mechanism of Tungsten Disulfide Lubrication in Vacuum

by M. T. Lavik, G. E. Gross and G. W. Vaughn

Midwest Research Institute, Kansas City, Missouri

The lubrication behavior of WS_2 in high vacuum has been investigated and found to be similar to that of MoS_2. The steady-state value for the coefficient of friction at room temperature is found to be 0.065. When sliding is resumed, following a "stop-time," the coefficient of friction is initially high. However, it drops rapidly and approaches the equilibrium value within a few minutes. The absorbed sulfur concept, hypothesized by Johnson for MoS_2 lubrication, is extended to explain the behavior of WS_2.

INTRODUCTION

The increased working temperatures and high bearing loads encountered particularly in the aircraft industry have resulted in lubrication demands which cannot be met by the more commonly known organic lubricants. This need has stimulated considerable research in the area of dry film lubrication, the most common example of which is the well-known graphite film. Because of this demand for special properties, many new materials, including graphite, molybdenum disulfide, tungsten disulfide have been tested.

In general the basis for selecting materials for test in this field has been that they have crystal structure similar to graphite or have low shear strength. In order to use logical methods in selecting materials for dry film lubrication, it is necessary that the basic physical mechanism of the phenomenon be understood. To meet this need for basic knowledge, several interesting research programs have been carried out. Perhaps the earliest of these was the work on graphite by R. H. Savage (1) in which it was established that the lubricant properties of the graphite film depend upon an absorbed layer of water vapor. Since his initial work, he has shown that certain other organic vapors may play the same role as water vapor (2). Evaporation of the adsorbed vapor film has been used to explain the failure of graphite as a lubricant in vacuum.

One material which does not suffer this characteristic failure in vacuum is molybdenum disulfide. Because of its well-known vacuum lubrication proper-

Presented at the 12th Annual ASLE Meeting in April, 1957, at Detroit, Michigan

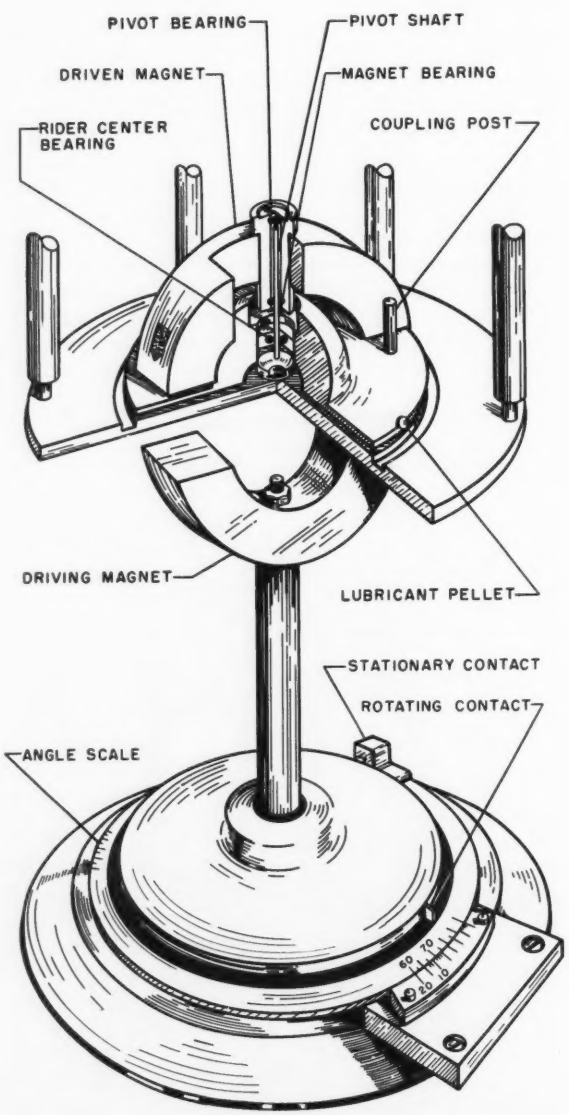

Fig. 1. Schematic view of friction test apparatus (from Ref. 4).

ties, V. R. Johnson (3) undertook an investigation of its lubrication mechanism. The results of Johnson's investigations indicate that the lubricant action of molybdenum disulfide is dependent upon an internally generated amorphous sulphur film which adheres to the crystal surfaces. Since two materials of laminar crystal structure had been so thoroughly investigated and found to indicate a vapor film lubricant mechanism, it was decided that the investigation should be carried to other laminar materials. Tungsten disulfide was then studied.

EXPERIMENTAL METHOD

Since the equipment used in this work is described in detail elsewhere (4), only a brief description will be given here.

A schematic diagram of the apparatus is shown in Fig. 1. The vacuum friction chamber is enclosed by a four in. diameter glass cylinder with O-ring seals on each end. The frictional track is formed on an elevated portion of the stainless steel base-plate and is lapped with 600 mesh carborundum prior to the testing of each sample. Three pellets of WS_2, compressed at approximately 60,000 psi, are inserted on the periphery of the brass pellet holder which provides the normal force. The pellets slide on the elevated frictional surface. Two Alnico-V magnets transmit the driving torque through the stainless steel (non-magnetic) base-plate. A pivot shaft, mounted in the base-plate, supports the weight of the driven magnet and also serves to center the pellet holder. Two posts mounted in the pellet holder couple it to the driven magnet.

The frictional force is a function of the lag angle beteween the two magnets, which is measured with a stroboscope. A direct calibration method is used to relate the lag angle to the frictional force. The instrumental uncertainty in the measured coefficient of friction is estimated to be approximately 0.005.

These experiments were conducted at a linear speed of 310 cm/sec with a load of 395 grams. The pressure within the friction chamber was maintained at approximately 1×10^{-6} mm. Hg unless otherwise specified in the discussion. With the exception of sulfur from the wear process or that added in one of the tests, the residual chamber atmosphere was air.

EXPERIMENTAL RESULTS

In all cases the films studied were "run-in" thoroughly before measurement to produce a WS_2 film on the frictional surface and to establish an equilibrium coefficient of friction. In general, the character of the results obtained is similar to those obtained by Johnson and Vaughn for molybdenum disulfide.

It is found that the coefficient of friction is above its equilibrium value when sliding is resumed after stopping for a given length of time. Fig. 2 shows the coefficient of friction for the first 60 minutes of running after stop times of from ½ hour to 18 hours with the film held at a temperature of 1C throughout running and stop time. Figures 3, 4, and 5 show similar results for temperatures of 24, 36, and 49C. It is seen in Fig. 3 that the initial coefficient of friction after ¼ hour of stop time at 24C is about 0.10, and at 24 hours' stop time, the coefficient has built up to an

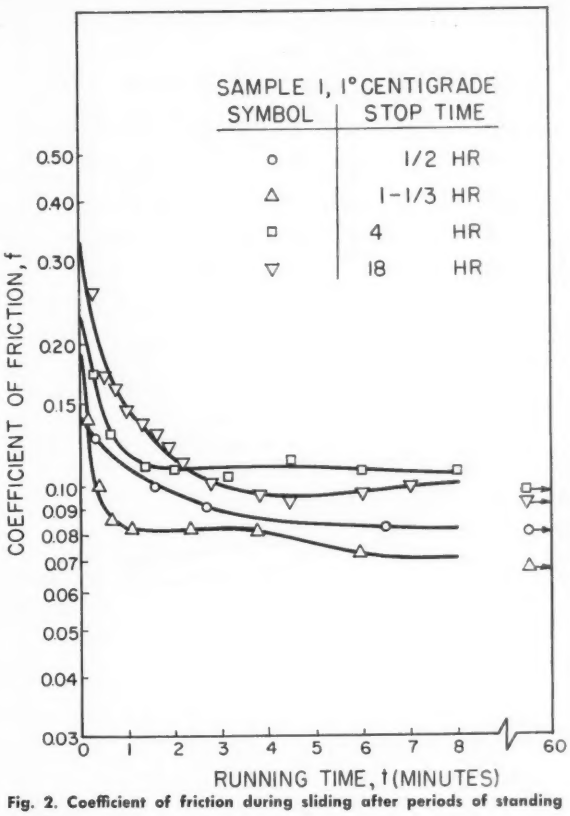

Fig. 2. Coefficient of friction during sliding after periods of standing at 1°C.

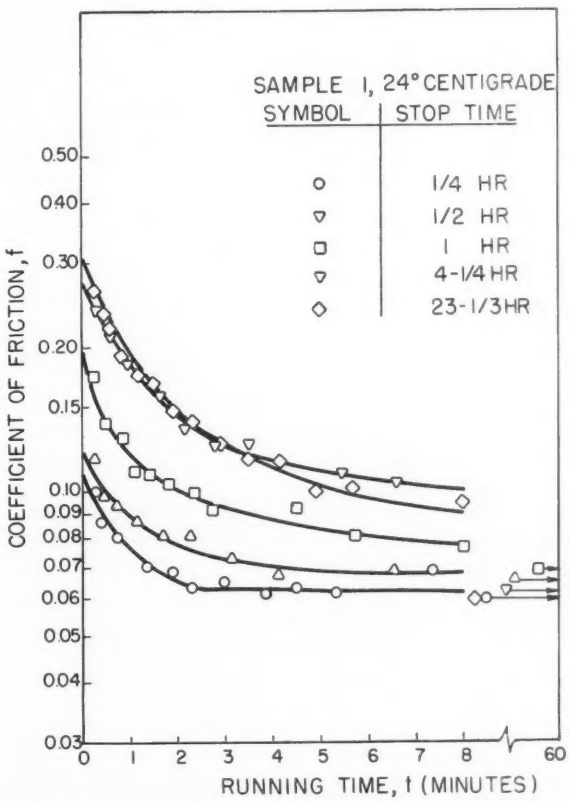

Fig. 3. Coefficient of friction during sliding after periods of standing at 24°C.

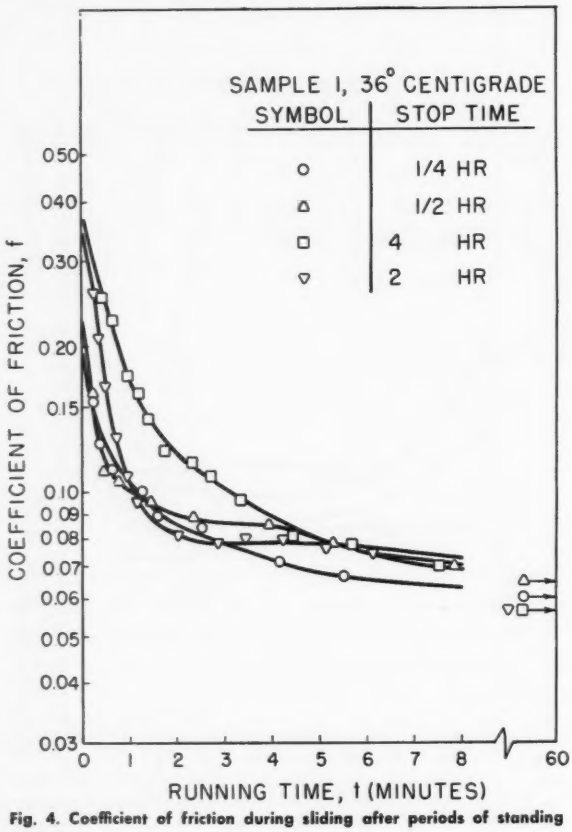

Fig. 4. Coefficient of friction during sliding after periods of standing at 36°C.

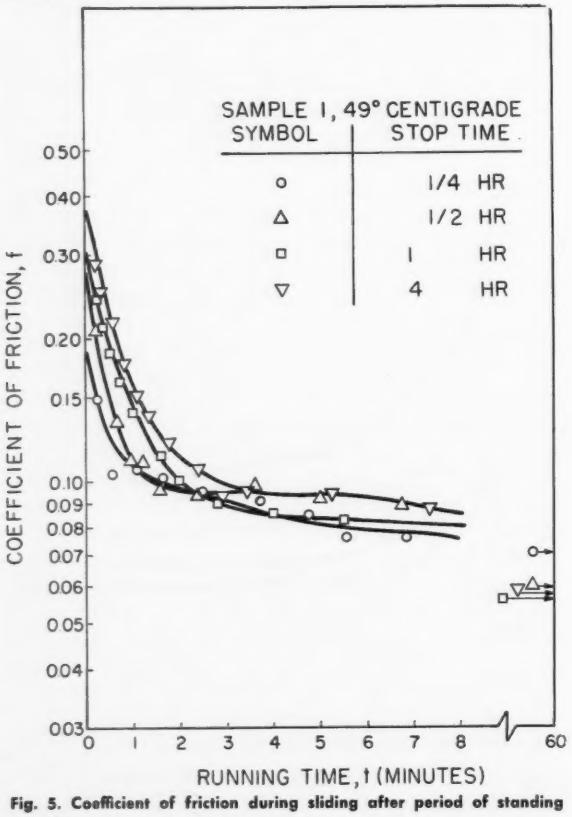

Fig. 5. Coefficient of friction during sliding after period of standing at 49°C.

initial value of 0.30. Experience showed that in all cases after running for 60 minutes, the coefficient has reached an equilibrium value. Fig. 5 shows that, after ¼ hour stop time at 49C, the coefficient has built up to an initial value of about 0.20 and that at four hours' stop time, it has built up to nearly 0.40. After 60 minutes' running, the final coefficient of friction was found to be about 0.065. This increased rate of friction buildup at higher temperatures suggests that an evaporation process may underlie the friction buildup. The equilibrium value of the coefficient of friction shows a small temperature dependence.

Fig. 6 shows the initial coefficients of friction (extrapolated values from Figs. 2-5) after various lengths of stop time at temperatures from 1° to 49C. For zero stop time, the initial coefficients (i.e., the equilibrium values) average about 0.065 and at stop times of 12 hours, the coefficients have an average of about 0.35. The rate of friction buildup shown by the initial slopes of the curves in Fig. 5 are seen to increase with temperature.

Qualitative observations showed that the rate of coefficient buildup was noticeably lower when sulfur vapor was introduced into the friction chamber. Quantitative measurements were not possible on the apparatus as used here and thus this observation was not confirmed in detail.

The buildup of the coefficient of friction f during stop time suggests an evaporation phenomenon and the run-in characteristic suggests a phenomenon which depends upon the sliding action itself. This behavior is similar to that described by V. R. Johnson (3) for molybdenum disulfide.

This hypothesis has been expressed approximately in mathematical form as follows:

A. For the period of coefficient buildup:

$$f_1 = f_o e^{-AGt_1} + f_c(1 - e^{-AGt_1}) \qquad [1]$$

where

f_1 = the initial value of the kinetic friction coefficient upon resumption of sliding.
f_o = friction coefficient for surface completely covered by amorphous sulfur.
f_c = friction coefficient for completely clean crystal surface.
G = evaporation rate of bulk sulfur.
A = a factor accounting for the difference in binding between the MoS_2 surface and bulk sulfur.
t_1 = time measured from cessation of sliding.

B. For the period of "run-in":

$$f_2 = f_0 + (f_c - f_0)e^{-(K/N_o)t_2} \qquad [2]$$

where

f_2 = the kinetic coefficient during run-in.
K = a constant.
N_o = number of adsorbed sulfur atoms per unit area for equilibrium lubrication.
t_2 = time measured from initiation of sliding.

The experimental results obtained here for WS_2 do not follow this description as well as desired and therefore a more detailed analysis has been undertaken. The new analysis considers the mechanical character-

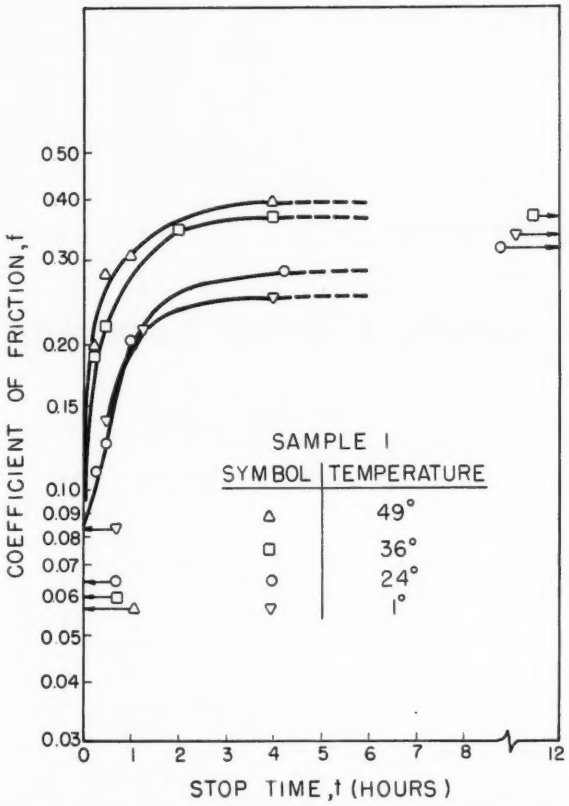

Fig. 6. Initial coefficient of friction as a function of stop time at various temperatures.

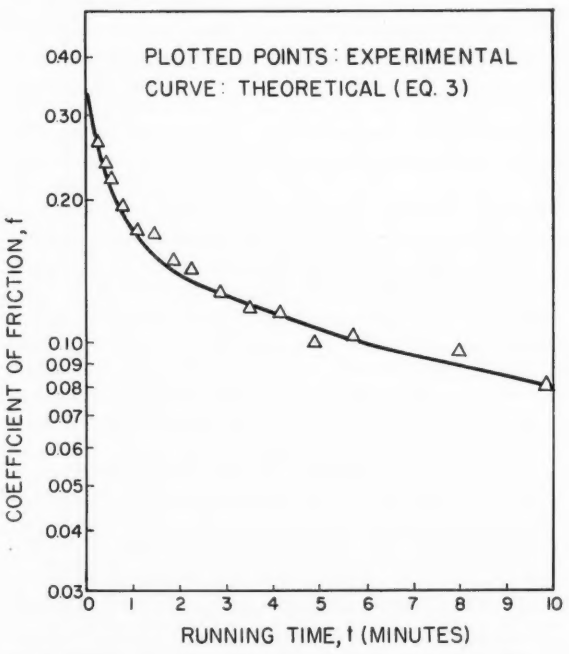

Fig. 7. Comparison of theoretical and experimental results for WS_2.

istics of the experimental setup and gives very close agreement with the data.

In the experimental setup as previously described, three pellets of the material under study are moved about a track upon which the pellets deposit a layer of the material. After the initial run-in period the frictional forces measured are those occurring between two surfaces of the material tested. The new analysis considers the following factors in addition to those already considered by Johnson:

(1) The test pellet and the track surface are produced by different mechanisms;—one by 60,000 psi pressure, the other by deposition in abrasion of the pellet. Thus the pellet should exhibit a different wear rate and a different sulfur buildup rate.

(2) The pellet is under constant abrasion while the track is abraded only when a pellet passes over any given point. Thus heating effects may differ and influence vaporation and deposition of sulfur.

(3) There must exist points of contact between sulfur on one surface and "clean" WS_2 on the other; in addition to the sulfur-sulfur and "clean crystal"-"clean crystal" contacts considered by Johnson.

These considerations result in a rather lengthy expression which may be approximated (within five per cent for t larger than 1.5 minutes) as follows:

$$f = f_{SS}(1 - e^{-A_T t}) + f_{SM}(e^{-A_T t} - e^{-BA_T t}) + f_{MM}\, e^{-(B+1)A_T t} \qquad [3]$$

where

f_{SS} = the coefficient of friction between sulfur films.
f_{SM} = the coefficient of friction of a sulfur film on WS_2.
f_{MM} = the coefficient of friction between WS_2 crystals.
A_T = the sulfur buildup rate coefficient for the track.
B = the ratio of the sulfur buildup rate of the pellet to that of the track.
t = time after initiation of sliding.

The values of the various coefficients and of the quantities A_T, and B were determined graphically from our data and the resulting curve is shown in Fig. 7. The experimental data are plotted on the same figure and the agreement seems to be very good. However, since the analytical curve has been fitted directly to the data, this agreement merely shows that the general type of mechanism assumed in describing the experiment is reasonable.

CONCLUSION

The adsorbed sulfur hypothesis was put forth by Johnson (3) in explanation of the friction "buildup" in MoS_2. This hypothesis has been modified to include certain experimental effects extant in the present work. The fact that the WS_2 data here agree with the modified hypothesis certainly suggests that the adsorbed vapor concept applies to this material.

There remains a lack of understanding concerning the details of the proposed adsorption process and its effect on the physical properties of the lubricant material. Therefore, it seems desirable to suggest an extension of the type of experiment reported by Campbell and Kozak (5) in which certain crystal orientations (of graphite) produced lubrication without adsorption vapors.

(Cont. on p. 264)

Some Factors Affecting Performance of Large Pivoted-Pad Thrust Bearings

by E. C. Kuhn and G. D. Cooper

Westinghouse Electric Corp., East Pittsburgh, Pa.

Oil-film thickness, pressure distribution, pad surface temperature, and pad surface distortion were measured on a large waterwheel generator thrust bearing during normal operation. These values were used to determine other characteristics of the bearing and compared with values predicted under the following conditions and assumptions:

- **a) Flat pad with constant viscosity and optimum pivot location**
- **b) Flat centrally-pivoted pad with variable viscosity and density**
- **c) Crowned pad with central pivot and uniform viscosity**

While the various assumptions used in predicting the performance give good agreement with test measurements for certain of the bearing parameters, no single set of conditions gives the desired degree of correlation for all parameters. On the basis of this test it is believed an analysis considering both pad curvature and variable viscosity would most accurately predict the performance of large centrally pivoted pad thrust bearings.

NOMENCLATURE

p—local oil-film pressure, psi
ϕ—pad polar coordinate
r—pad polar coordinate
h—local oil-film thickness, in.
μ—local oil viscosity, reyns
H—power loss, kw per pad
ω—angular speed, rad per sec
N—grid coordinate
M—grid coordinate
Q—lubricating oil flow, gpm per pad
$F(\ )$—function notation

Large thrust bearings required for high-capacity water-wheel generators impose unique problems in the design of the bearing. One of these problems is pad distortion due to load and thermal gradients and its effect on hydrodynamic performance of the thrust bearing.

In the design of pivoted-pad thrust bearings, it is conventional to assume plane surfaces and uniform viscosity of the lubricating oil film. The use of treatments of Boswall (1), Muskat, Morgan, and Meres (2), and Raimondi and Boyd (3), to cite a few, have enabled good predictions of the performance of bearings with plane surfaces and pivot offset from center. It is mandatory in applying these theories with uniform viscosity that the pivot be positioned downstream from the geometric center of the pad. In many applications pivoted pads have been used with the pivot located at the geometric center of the pad, e.g. to obtain reversibility of the direction of machine rotation. The fact that centrally-pivoted pads operate satisfactorily indicates that flat-pad theory with constant viscosity is not adequate to handle the central pivot case.

The change in viscosity of the lubricating oil with temperature and pressure as well as a decrease in density with temperature are factors which affect load capacity of centrally-pivoted pads. A paper by Charnes and Saibel (4) considers the increase in viscosity with pressure. This effect alone does not appear to explain the load capacity of centrally-pivoted pads. According to Norton (5) and others, substantial load capacity of centrally-pivoted pads can be realized considering temperature variation of viscosity. The decrease in density with temperature according to Shaw (6) has an effect on bearing performance. As well as these factors, surface profile of the bearing surfaces affect thrust bearing performance. In a treatment by Ying, Charnes, and Saibel (7) transverse curvature is shown to decrease the load capacity of slider bearings. Elastic deformation of bearing surfaces and its effect on performance was considered by Osterle and Saibel (8). An analysis by Raimondi and Boyd (9) indicates that surface crown or convexity contributes substantially to the load capacity of centrally-pivoted pads.

Distortion of rectangular pads in the direction of motion and transversely has been investigated by Raimondi (10). This analysis assumes constant viscosity and employs a digital computer program. Separate solutions are available for distortion in the direction of motion, in the transverse direction and for spherical curvature (equal in both directions). It was

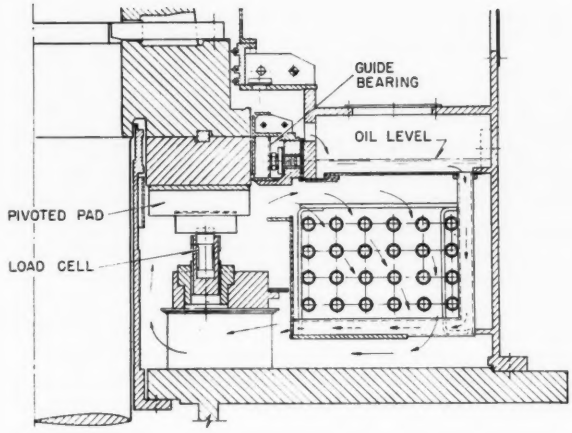

Fig. 1. Sectional view of large waterwheel generator thrust bearing of type tested.

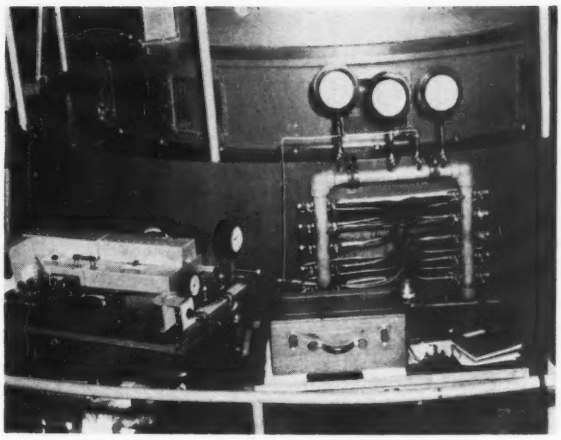

Fig. 2. Arrangement of test equipment near thrust bearing during actual field testing.

shown in this treatment that an optimum amount of crown in the direction of motion permits the establishment of an adequate oil film. In the case of spherically-crowned pads, the load capacity of the centrally-pivoted pad was shown to be equivalent to the load capacity of the hypothetically flat pad with optimum pivot location.

Large waterwheel generators have been designed with centrally-pivoted pad thrust bearings for many years. The satisfactory operation of this type of thrust bearing has been assured as a result of extensive laboratory tests together with field investigations. The excellent performance of these bearings has been attributed to a desirable combination of variation in viscosity and surface curvature due to load and thermal gradients in operation. As was discussed by Baudry, Kuhn, and Wise (11) distortion of the pad surface must be maintained at a low value to achieve optimum performance of centrally-pivoted pads during normal operation.

In the case of very large bearings (8 ft dia) with capacities of over 1 million lb, laboratory tests are impractical. In such cases tests on actual bearings in operation yield the most valuable information toward a better understanding of the factors involved in bearing design. Recently certain tests were made on the thrust bearing of a large waterwheel generator during normal operation. This article will discuss a portion of the data obtained during that field test. A paper by Baudry, Kuhn, and Cooper (12) describes the tests and gives a description of instrumentation developed for this investigation.

DESCRIPTION OF THE BEARING AND MEASUREMENTS

The thrust bearing upon which these data were obtained was 96 in. dia and consisted of 8 sector-shaped pads each centrally-pivoted and supported by individual jack screws. As shown by Fig. 1, each jack screw was provided with a calibrated load cell for the purpose of adjusting the bearing load equally among the pads. The normal operating speed of the bearing was 120 rpm. The observed bearing load at maximum electrical output was 1,110,000 lb. One of the pads was instrumented to measure:

a) Oil-film thickness
b) Oil-film pressure
c) Pad surface temperature

Fig. 2 shows the arrangement of test equipment near the thrust bearing during field testing.

RESULTS AND DISCUSSION OF RESULTS

a) *Oil-Film Thickness and Pad Deformation*

During normal operation, oil-film thickness (the distance from runner surface to pad surface) was measured at 14 locations over the pad surface by .625 in. dia electro-magnetic gages. Change in reluctance of the gage circuit, which included the oil-film and runner, was observed. These values of film thickness were used to develop the surface shown by Fig. 3 which represents the pad surface to an exaggerated scale. Fig. 4 shows the oil-film shape and displays the relative positions of the pad surface and runner during normal operation. The observed film shape along the pad's circumferential center line and the film shapes

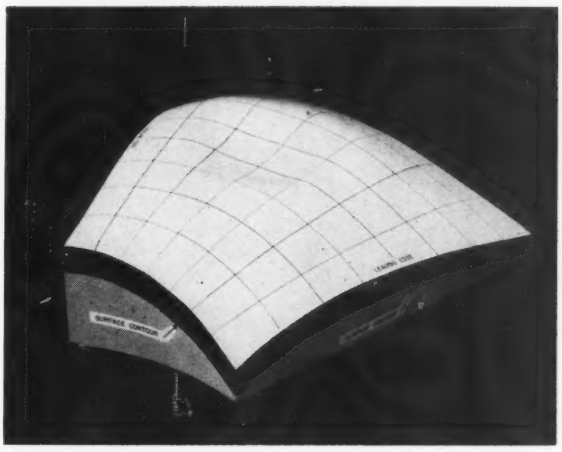

Fig. 3. Model of thrust bearing pad showing measured surface deformation during normal operation at 120 rpm. Measured pad load was 139,000 lb.

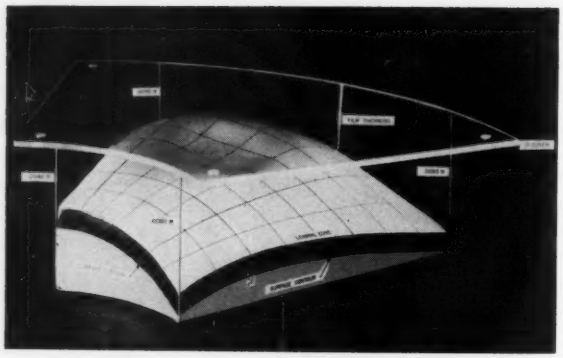

Fig. 4. Display showing observed oil film shape and relative position of the thrust pad and runner during normal operation.

predicted by theories with various assumptions are shown on Fig. 5. The observed film shape is convergent up to a point of minimum film thickness at some distance from the outlet edge of the pad. The film shape becomes divergent between this region and the outlet edge. As is apparent from Figs. 3 and 4, the pads of this particular bearing had scraped into their centers a circular relief area about .001 in. deep. This was in accordance with established practice at this installation. Such a central relief adds considerable complication in applying available analytical solutions. It is believed reasonable correlation does exist by neglecting this recess in applying available analytical treatments. The actual surface is considered here.

b) *Hydrodynamic Pressure in Oil Film*

The oil-film pressure was measured at 20 locations over the pad surface by small-bore copper tubing emanating from the oil film. Readings were obtained by pressure gages. These values were used to develop the model of pressure distribution shown by Fig. 6. The integral of the pressure, determined by weighing the model, was found equal to the pad load of 139,000 lb as read by the resistance wire strain gage located in the pivot of the pad. This model shows the pressure to be distributed over the pad with a peak of about 700 psi near the pivot location. Applying flat-pad theory with pivot positioned for maximum load capacity and considering a linear variation of viscosity in the direction of motion, the pressure distribution was calculated for the test bearing. In this case the maximum pressure was about 525 psi. Fig. 7 shows a comparison between the measured and hypothetical pressure distributions. Fig. 8 shows these pressure distributions along the circumferential paths passing through the location of maximum pressure for each case. As is noted for the actual pad, the pressure distribution is more concentrated at the center of the pad and of greater maximum value than predicted by flat-pad theory.

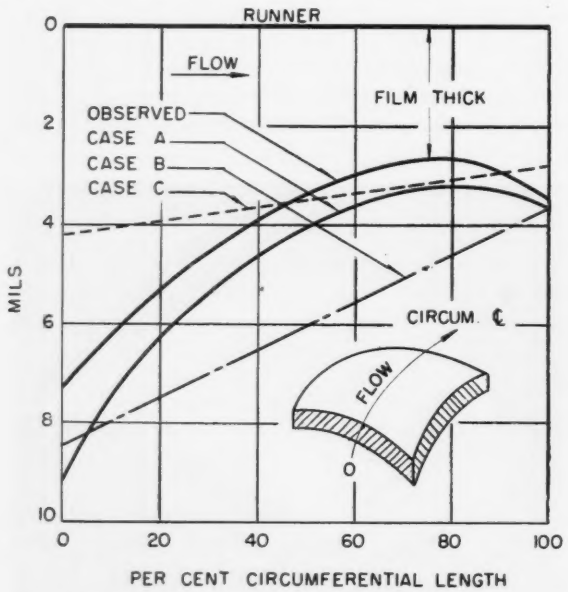

Fig. 5. Film thickness and pad surface deformation along pad's circumferential centerline as measured, and film shapes predicted by:
A—Crowned-pad, uniform viscosity, central pivot
B—Flat-pad, uniform viscosity, optimum pivot
C—Flat-pad, variable viscosity, central pivot

c) *Pad Surface Temperature and Viscosity Distribution*

Temperatures were obtained by use of copper-constantan thermocouples mounted in series of five into a .625 in. dia plug pressed into each of six reamed holes perpendicular to the pad surface. Surface temperatures were ascertained by extrapolating these temperature values. The arrangement of thermocouples facilitated predicting surface temperature along the pad's radial centerline and a circumferential line near the center of the pad. Fig. 9 shows a plot of these observed temperatures.

In determining the viscosity of the oil it was assumed that the film temperature was equal to the observed surface temperature. It was further assumed that the viscosity everywhere in the plane of the bearing could be expressed as a product of the viscosities along these two lines, i.e.

$$\mu_{r\theta} = F(\mu_\theta)F(\mu_r)$$

From Table 1, which was prepared from the data, values of viscosity were determined for each of 49 dis-

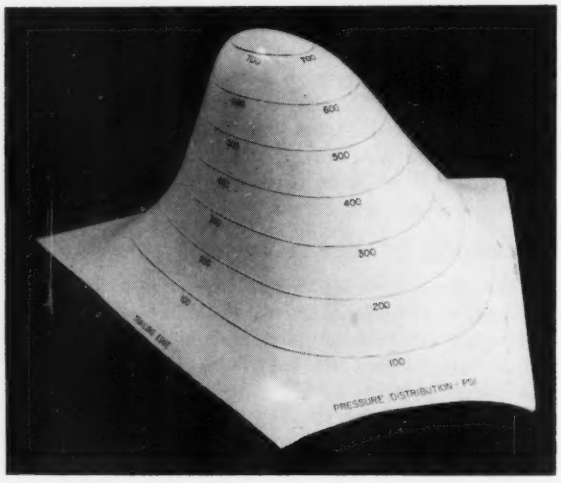

Fig. 6. Model of observed oil-film pressure distribution during normal operation at 120 rpm with pad load of 139,000 lb.

TABLE 1. VISCOSITY FUNCTIONS

PER CENT LENGTH	$F(\mu_r)$ DIMENSIONLESS	$F(\mu_\theta)$ CP.
0	1.41	31.0
10	1.33	27.6
20	1.23	24.6
30	1.13	22.0
40	1.07	20.0
50	1.02	18.4
60	1.00	17.0
70	1.00	16.0
80	1.02	15.2
90	1.04	14.5
100	1.09	14.6

crete points over the pad surface. These positions are shown on Fig. 10 as the intersection of the heavy grid lines and the edges of the pad.

OTHER BEARING CHARACTERISTICS

Characteristics of a bearing such as minimum oil-film thickness, power loss, oil flow and side leakage, and film temperature rise are usually estimated by the bearing designer. It is convenient to show additional characteristics for the bearing tested if using observed data together with theoretical equations will be considered a suitable substitute for the characteristics not easily measured.

a) *Power Loss*

From the model shown in Fig. 6, values of pressure were ascertained at 133 discrete points over the pad surface by probing the model in the manner suggested by Fig. 11. The locations of these pressures as well as the corresponding film thickness (obtained from the film-shape model Fig. 4) are shown on Fig. 10. The power loss of the pad was calculated by graphically determining the pressure gradients in the direction of rotation at the discrete points;

$$\begin{array}{l} N_3M_1\ ,\quad N_3M_5\ ,\quad N_3M_7\ ,\cdots N_3M_{13} \\ N_5M_1\ ,\qquad\qquad\qquad\qquad\quad \cdots \\ N_7M_1\ ,\qquad\qquad\qquad\qquad\quad \cdots \\ N_9M_1\ ,\qquad\qquad\qquad\qquad\quad \cdots \\ N_{11}M_1\ ,\cdots\cdots\cdots\cdots\cdots\cdots N_{11}M_{13}\ , \end{array}$$

shown on Fig. 10.

Power can be expressed as:

$$H = \int_\theta \int_r \left[\frac{\mu\omega^2 r^3}{h} + \frac{\omega r h}{2}\left(\frac{\partial p}{\partial \theta}\right)\right] dr\, d\theta.$$

This integral can be solved numerically by the summation,

$$H = \Delta r\, \Delta\theta \sum_{N=3,5,7}^{11} \sum_{M=1,3,5}^{13} \left[\frac{\mu\omega^2 r^3}{h} + \frac{\omega r h}{2}\left(\frac{\partial p}{\partial \theta}\right)\right] + \frac{\Delta r\, \Delta\theta \omega^2}{2} \sum_{M=1,3,5}^{13} \left[\left\{\frac{\mu r^3}{h}\right\}_{N=1} + \left\{\frac{\mu r^3}{h}\right\}_{N=13}\right].$$

b) *Oil Flow and Side Leakage*

The flow of oil in the direction of rotation is composed of two flows; shear flow and pressure gradient flow. These can be expressed as

$$Q = \int_r \left[\frac{\omega r h}{2} - \frac{h^3}{12\mu r}\left(\frac{\partial p}{\partial \theta}\right)\right]$$

dr, and solved numerically by the summation

$$Q = \frac{\Delta r}{2} \sum_{N=3,5,7}^{11} \left[\omega r h - \frac{h^3}{6\mu r}\left(\frac{\partial p}{\partial \theta}\right)\right] + \frac{\omega\, \Delta r}{4}\left[rh\right\}_{N=1} + rh\}_{N=13}].$$

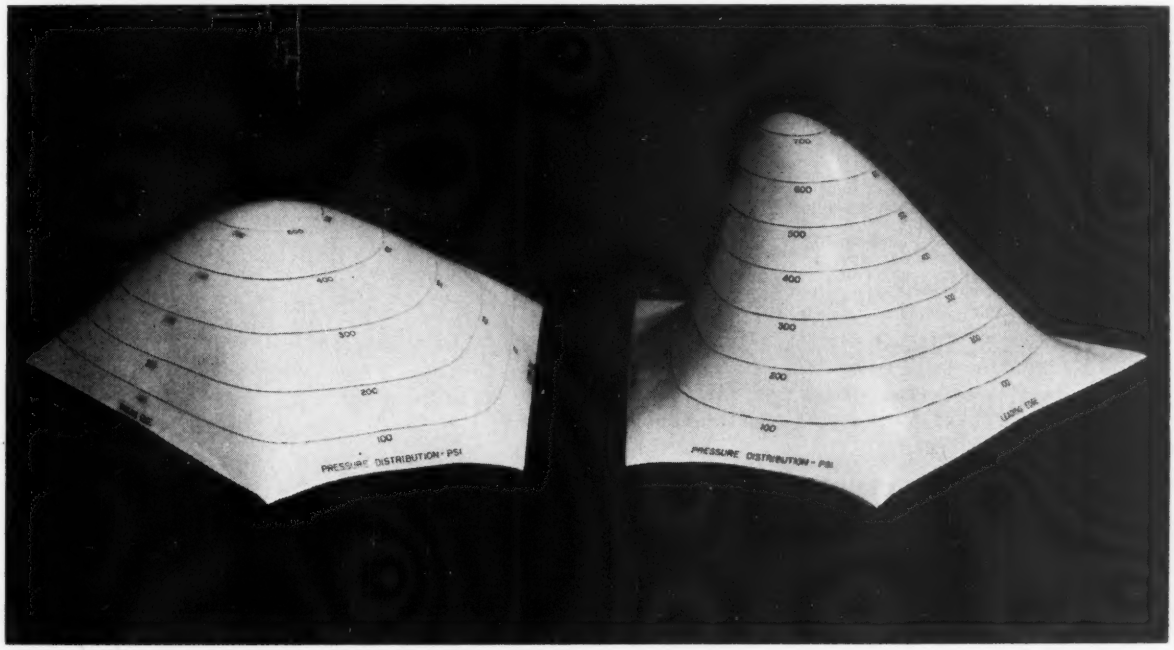

Fig. 7. Models showing comparison of oil-film pressure distribution during normal operation. Left, pressure distribution considering flat-pad theory with pivot offset for maximum load capacity and variable viscosity in the direction of motion. Right, observed pressure distribution for this centrally-pivoted pad.

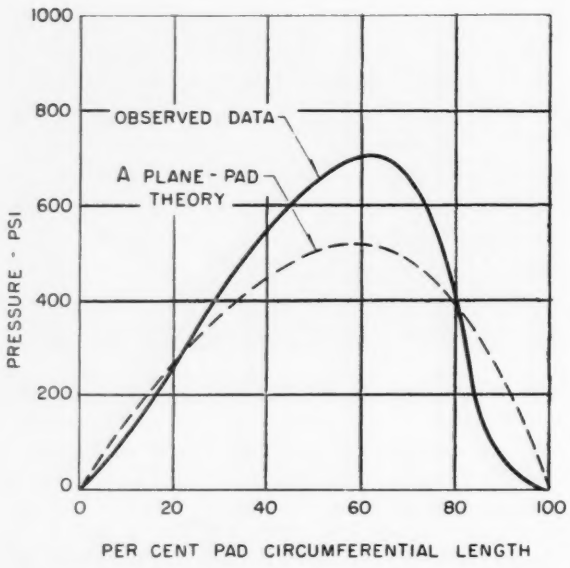

Fig. 8. Comparison of pressure distributions along circumferential lines passing through positions of maximum pressure. Flat-pad theory considers variable viscosity in the direction of motion with pivot offset for maximum load capacity.

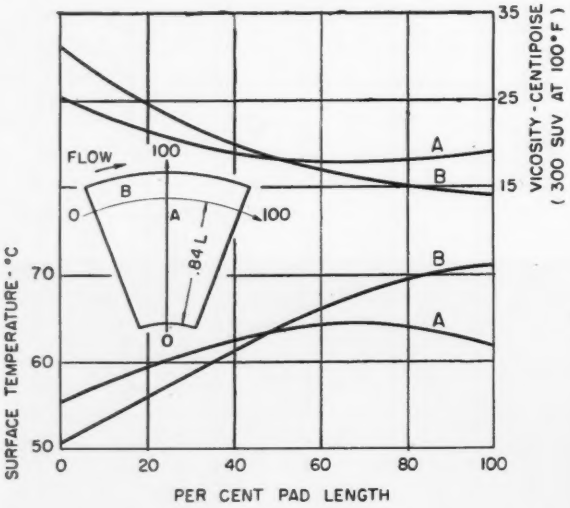

Fig. 9. Pad surface temperatures and viscosities observed during normal operation at 120 rpm and with pad load of 139,000 lb.

The oil flow rate for the pad was evaluated at the leading edge ($M=1$) and at the trailing edge ($M=13$). It was considered that the difference between these values of flow was side leakage.

OBJECT AND RESULTS OF ANALYSES

The object of this article has been to present some measured information on the hydrodynamic performance characteristics of a large centrally-pivoted thrust bearing pad during normal operation in the field and to compare these values with those predicted by theories pertaining to the satisfactory performance of centrally-pivoted pads.

Table 2 shows the characteristics of this bearing

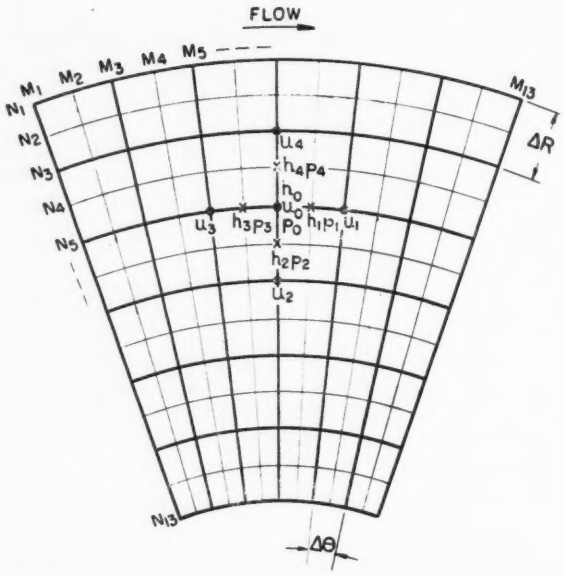

Fig. 10. Grid used for numerical computations of bearing characteristics. 133 discrete points were used in the analysis of power loss and oil flow.

predicted by flat-pad theory considering variable viscosity and density in the direction of motion. This analysis was made from unpublished data of Raimondi and considers simultaneously the Raynold's and energy equations taking into account variation in viscosity and density due to temperature.

Also shown are values predicted by Boswall (1) for this bearing with hypothetically plane surfaces, uniform viscosity, and pivot offset for maximum load capacity.

In view of the observed curvature of the pad, it seems far from reality to neglect surface profile and its effect on the performance of pivoted-pad thrust bearings. Raimondi and Boyd (9) and Raimondi (10) show surface crown of magnitudes similar to those ex-

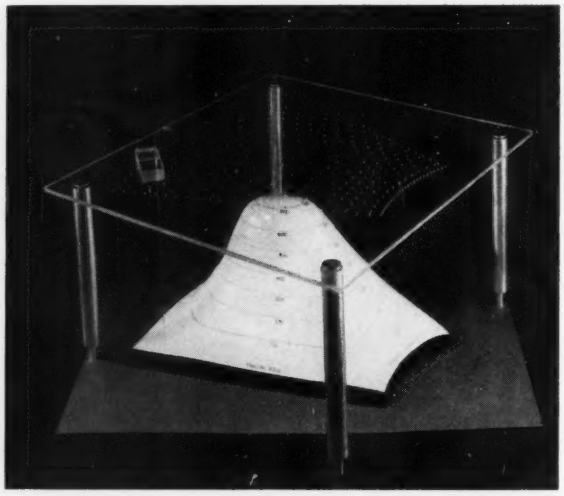

Fig. 11. Display showing method used to determine discrete values of pressure and oil-film thickness by models prepared from observed data.

TABLE 2. RESULTS

PERFORMANCE CHARACTERISTIC	FLAT PAD		CROWNED PAD	TEST
	VARIABLE VISCOSITY CENTRAL PIVOT	CONSTANT VISCOSITY OPTIMUM PIVOT	CONSTANT VISCOSITY CENTRAL PIVOT	
Min. Film Thick.-in.	.00282	.00368	.00320	.00270
Inlet Film thick.-in.	.00420	.00845	.00921	.00730
Outlet film thick.-in.	.00282	.00368	.00366	.00345
Ratio-inlet to outlet	1.49	2.30	2.52	2.12
Longitude crown-in.	0	0	.00190	.00190
Avg. temp. rise-C	20.2	10.5	11.4	19.4
Power loss-kw	12.4	10.4	11.6	13.3
Flow-gpm	6.11	11.00	11.10	11.45
Side leakage-gpm	1.08	3.96	3.18	3.70

Oil Properties: 8.16×10^{-6} reyns at 100 F, $.800 \times 10^{-6}$ reyns at 210 F. Sp. Heat = .475 btu per lb, density = .0308 lb per in.^{3}

Comparative analysis based on oil at same inlet temperature.
Spherical crown = 2 × longitudinal crown

perienced during this test to be of primary significance in contributing to the load capacity of centrally-pivoted pads. Table 2 shows the characteristics of this bearing predicted by crowned-pad theory using the measured value of crown along the circumferential center line of the pad and uniform viscosity.

The hydrodynamic characteristics of the actual bearing at maximum load and normal speed are also shown in Table 2. The test data show that an adequate oil film exists with moderate rise in temperature.

CONCLUSIONS

1. The test data indicates that this centrally-pivoted pad is operating very satisfactorily with an adequate oil film and moderate oil-film temperature rise.
2. Predictions by crowned-pad theory are generally in good agreement with values obtained from this test. In the case of flat-pad theory with variable viscosity and density some parameters are in good agreement with test measurement. As a result of this test it is believed analyses considering both crown and variable viscosity would most accurately predict the performance of large centrally-pivoted pad thrust bearings.

ACKNOWLEDGMENT

The authors wish to thank Mr. A. F. Darland at Coulee Dam for the opportunity to conduct these tests initiated by Mr. R. A. Baudry of Westinghouse Electric Corporation. We are grateful to Mr. John Boyd and Mr. A. A. Raimondi of the Westinghouse Research Laboratories for their comments in the preparation of this paper.

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Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

"A Theory of Lubrication in Short Journal Bearings with Turbulent Flow," by L. N. Tao, Trans ASME v 80, n 8, 1958, pp 1734-40.

For bearings operating under conditions where flow in the oil film is turbulent, the well-known Sommerfeld analysis does not agree with operating results. An analysis of the bearing characteristics of bearings with turbulent lubricant flow is developed. Equations are developed for oil film pressure, load capacity and attitude angle. Application of the method of analysis and comparison of the results with other methods and with test results are discussed. (*Abstracted by G. B. Grim*)

"Laboratory Evaluation of Fire-Resistant Turbine Fluids," by G. H. S. Snyder (Socony Mobil Oil Co., Paulsboro, N. J.), L. W. Manley, and N. V. Messina, ASTM Special Technical Publication, n 211, 1957, pp 5-17.

Types of fluids and their evaluation are reviewed. Tests are discussed for flammability, oxidation and hydrolytic stability, corrosion, foaming and demulsibility properties, effects on seals and insulating materials, and lubricating qualities. (*Abstracted by M. I. Smith*)

"Antiwear Requirements for Navy Turbine Oils," by H. F. Kings (Bureau of Ships, Navy Dept., Washington, D. C.) and J. R. Belt, ASTM Special Technical Publication, n 211, 1957, pp 18-39.

The laboratory evaluation program described was prompted by recent experiences with scuffing of turbine gears aboard naval vessels. Fourteen test methods are investigated. Although the reproducibility of the Ryder Test leaves something to be desired, it is used as the basic qualification test, calling for a minimum of 2200 lbs at 22½ per cent scuff rating. The 4-Ball mean Hertz load and wear tests are used as further safeguards. Radioactive wear tests give only a rough correlation with the Ryder rating. The latter is found to depend upon the configurations of the test gears. (*Abstracted by M. I. Smith*)

"Rusting in Turbine Oil Systems," by N. W. Furby (California Research Corp., Richmond Cal.), F. J. Hanly, and J. A. Vincent, ASTM Special Publication, n 211, 1957, pp 49-54.

Fundamental mechanisms of rusting are reviewed. Rust inhibitors function by increasing the metal-wetting ability of oil relative to water. They do not protect surfaces above oil level, nor can they indefinitely protect horizontal surfaces on which water has settled. These factors are not revealed by ASTM Rust Test results. It is recommended that moisture be removed as efficiently as possible by a "positive venting" system. A method of monitoring actual systems is suggested whereby test panels are immersed in the oil and vapor spaces, and set on the tack bottom. (*Abstracted by M. I. Smith*)

"Monomolecular Films of Rust-Preventive Additives," by H. E. Ries, Jr. (Standard Oil Co. of Ind., Whiting, Ind.), H. D. Cook and C. M. Loane, ASTM Special Technical Publication, n 211, 1957, pp 55-68.

Performances of polar additives in ASTM rust and film tenacity tests are compared with fundamental studies of thin film properties. These studies were made with the aid of the film balance, electron microscope and radiotracers. Film structure is correlated with rust-preventive ability. Water is found to increase the effectiveness of rust inhibitors, probably as the result of oxide formation and roughening of the steel surface. (*Abstracted by M. I. Smith*)

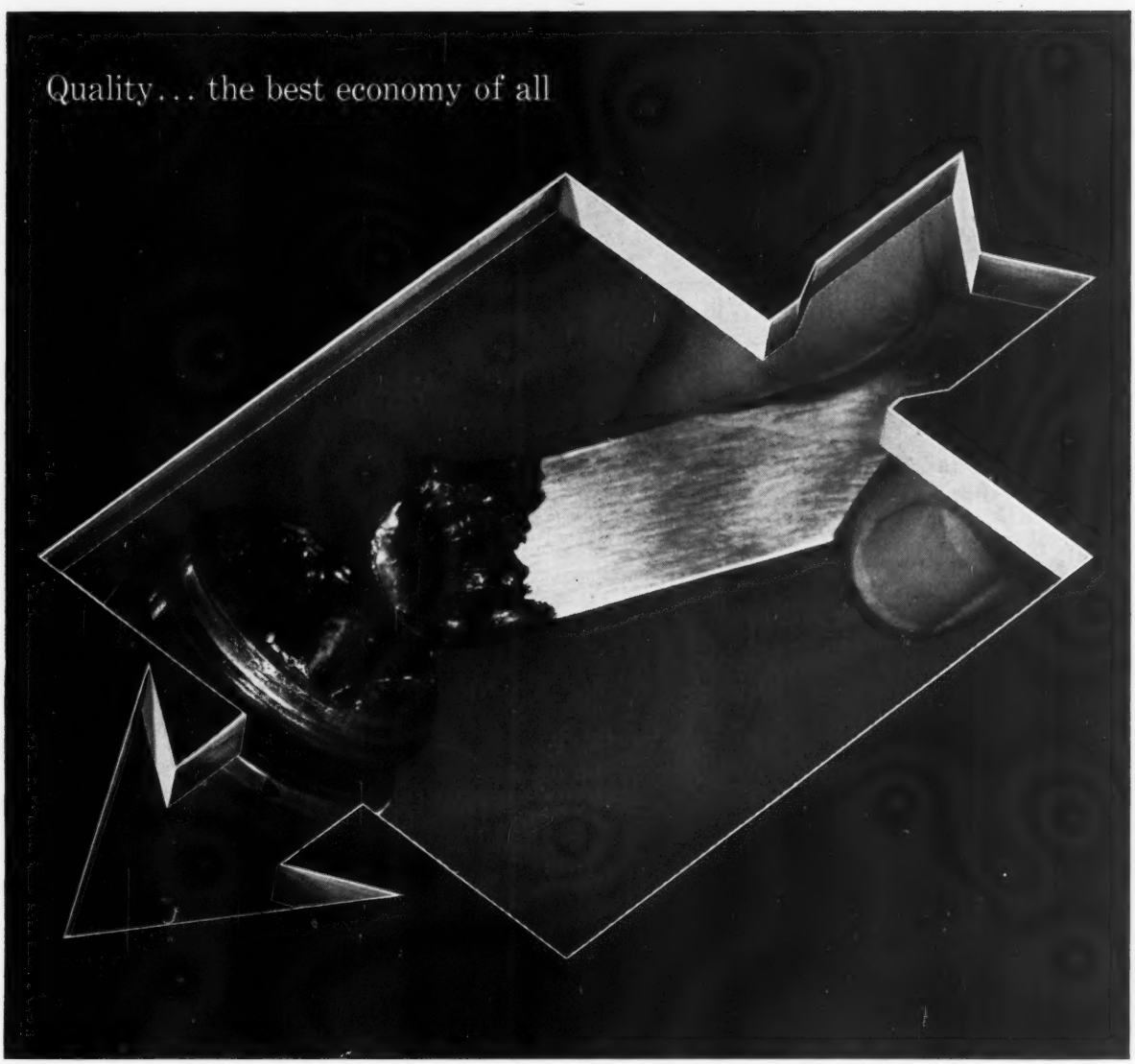
"A Laboratory Device for the Water Leaching of Additives in Turbine Oils," by E. W. Brennan (Pure Oil Co., Crystal Lake, Ill.), and R. G. Moyer, ASTM Special Technical Publication, n 211, 1957, pp 69-75.

Equipment previously described for the authors' Dynamic Demulsibility Test is used to contact turbine oils and water, after which performance tests are conducted on the separated oil. (*Abstracted by M. I. Smith*)

"Practices for Determining the Expected Life of Used Turbine Oils," by R. G. Mastin (Cities Service Oil Co., E. Chicago, Ind.), ASTM Special Technical Publication, n 211, 1957, pp 76-92.

Case histories of turbine oils in service over a period of ten years are given. Addition of rust inhibitor is often required, the concentration to be added being determined by interfacial tension measurements. Clay treatment to restore oxidized oils to good condition is seldom needed. Improvement of used oils by addition of anti-oxidant is recommended when A.S.T.M. T.O.S.T. life falls below 250 hours. (*Abstracted by M. I. Smith*)

(*Cont. on p. 262*)



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ASLE Technical Digest

Papers which, because of space limitations, cannot be published in full in Lubrication Engineering or ASLE Transactions.

"Steel Mill Lubrication, The Value of Records," by Lewis J. Reguly, Inland Steel Co., Chicago, Ill.

Lubrication iron and steel mill machinery has come a long way since the traditional oil can and prototype of the modern grease gun. Speed-up, increased load and advancements in bearing, seal and gear design all dictated the need for comparable modernization of the lubrication procedure. Today this is accompanied by specialty lubricants designed to function most dependably under the prevailing heavy-duty service—service which may require large generating units to operate for months, even years without stopping. Performance is the measure of the suitability of the lubricant, just as it is for the machine. Physical tests can only go so far in denoting what can be expected from the lubricant. Lubricants in steel mills must function under a wide variety of adverse conditions, the worst of which involve scale and dirt. Blast furnace dust, coal and cinder breeze, iron and slag particles together with gas fumes are constantly surrounding and sifting into lubricating systems. So, instead of having a pure, clean and stable lubricant as supplied by the manufacturer, a grinding compound may result.

Records, case histories and cost data all must be available in order to evaluate performance of the machine, the lubricating system and the lubricant. Significantly, data favorable to one will normally be favorable to the others. In an industry such as steel, where so many varieties of lubricants may be required, and where the lubricating systems are becoming so complicated and positive in action, performance data is a must.

Data is most dependable when accumulated by informed personnel. This accounts for the practice of properly notifying all interested parties as to types of lubricants, the reasons for their selection and the basic physical characteristics. Later, if changes in formulation are considered advisable by the lubrication engineer in consultation with management, the same personnel is informed as to the reasons for the changes. This breeds a spirit of confidence in the people actually responsible for application of lubricants and maintenance of lubricating equipment. They will be more disposed to accept the advice and information from supervisors when the personal ability and competence of the latter is recognized and respected. Table 1 is a typical information bulletin.

TABLE 1. Departmental Bulletin

SECTION. Lubricants.	DATE	PAGE	NUMBER
SUBJECT Improved Compounds Inland Code No. L-1001 & L-2601	ISSUED BY Sr. Mechanical Engineer Mechanical Dept.		
DISTRIBUTION			

Please be informed that improvements have been announced by the supplier to the above subject compounds.

In order to take advantage of this advanced formulation, we have approved of this change.

Hereafter these compounds which are compatible with the compounds you are now using, will be carried in Stores stock. They have the following specifications:

CODE NO.	*L-1001*	*L-2601*
Flash	360	400
Vis. @ 100F	1350	2400
Vis. @ 210F	95	129
Pour Point	0	10
Color	Dark Green	Dark Green

Both of these have the following characteristics:

1. High load carrying ability.
2. Good chemical stability, even at high operating temperature.
3. Ready separation from water.
4. Non corrosive.
5. Excellent rust protection.
6. Maximum wear resistance.

Please have authorized personnel note and initial this bulletin.

Approved:

Superintendent
Mechanical Department

Cost of lubrication, broken down by product type, number or code, and department, is an important item in the record keeping program. Along with cost of lubrication, performance comes into the picture. Faulty lubrication may be due to a product being off specification or to some other variable of a more mechanical nature. Some of the points to be checked and questioned include:

TABLE 2

LUBRICATION RECOMMENDATIONS				DEPARTMENT UNIT FOREIGN NO. INLAND NO.
DESCRIPTION:				
PART	METHOD	LUBRICANT SUMMER / WINTER		INTERVAL

1. Has mill operation changed? It faster, harder or colder steel being rolled? All may increase bearing pressures.
2. Are all bearings getting the recommended clean lubricant at the proper interval and in the right amount?
3. Is the babbitt metal up to specification and are correct pouring procedures being followed?
4. Are the journals in good condition? They may be spalled, rusted, nicked, undersize, eccentric or improperly finished.

Complete report as to basic cause, when finally submitted, is a valuable adjunct to the files of the lubrication engineer and the mechanical people. It becomes a case history which can serve as a warning against repetition or as a good example of what should be done if similar trouble recurs.

In the maintenance of steel mill lubrication, some attendant mechanical maintenance must always be going on. Records of such maintenance, in the form of monthly progress reports showing the status of the various jobs, are most informative to management as a part of the annual cost of operations. Correctly designed and lubricated machinery will greatly reduce maintenance costs. Proper lubrication, regular attention and systematic cleaning during service, as well as proper choice, handling and installation of replacements, together with adequate records, are most important in attaining normal bearing life, full production, low maintenance and low production costs.

On the other hand damage can sometimes be caused lubrication appliances on mill equipment, which can be repaired only on a scheduled "down" turn. As a result, good usage is made of reports which show when each mill is operating and when it is scheduled "down." This information is very helpful because it permits the maintenance crews to plan their repair jobs ahead of time.

To insure that new equipment is properly lubricated, a lubrication specification is included with all bids for new machinery, with information on (a) parts to be lubricated, (b) frequency of lubrication, (c) method of lubrication, and (d) lubricants recommended for winter and summer. This results in complete cooperation between the machinery builder and the lubrication department. special lubrication recommendation form This information then is compiled on a (see Table 2) which is distributed to the erectors of the equipment, the mechanical foreman in charge of maintenance, and to each maintenance man.

(*Digested by A. F. Brewer*)

"Soluble Oils and Soluble Cutting Lubricants," by M. C. Burak, Macco Prod. Co., Chicago, Ill.

Progress in the art of metal cutting using water-soluble materials for heat removal and lubrication has been paralleled by progress in the study of the available materials. Water was the original cutting fluid. Its usage increased cutting speeds from 30 to 40 per cent, but the natural tendency to promote rusting of tool and work was objectionable. To reduce this tendency, sodium carbonate or bicarbonate solutions in water were substituted. To some extent, these also increased the lubricating ability. Research involving various soaps and subsequent development of soluble oils was prompted by paint removal and by skin infections. Soluble cutting lubricants are the latest developments. They differ from the soluble oils, which are basically coolants, because they have added extreme pressure qualities, anti-weld ability and a degree of oiliness which greatly improves the lubricating ability. Like the soluble oils, soluble cutting lubricants are prepared to reduce the dermatitis possibility.

These products involve the basic theory of emulsification. Specifically the emulsifying agent must be compatible with the oil to be emulsified. This emulsifier must be a compound composed of molecules which are oil-soluble at one end and water-soluble at the other. Petroleum sulfonates, fatty acids or amines have this characteristic.

All soluble oils and soluble cutting lubricants should have certain definite characteristics if they are to function successfully in cutting modern high alloy steels. i.e.

1. Emulsion stability regardless of type of water.
2. Absence of gum deposition on the machine.
3. Maximum non-rusting tendency, either on machine, tool or work.
4. Non-foaming.
5. Freedom from rancidity or objectionable odors.
6. Lubricating and cooling ability so that the metal-removing job can be done dependably.

This latter objective can be assured most positively when the emulsion is permanent. The best practice is to add a soluble oil or soluble cutting lubricant to water while the latter is being agitated, at the point in the tank or mixing container where most active agitation is occurring. Two basic reasons require this procedure:

1. If oil is added prior to water, it will tend to stick to the sides of the container and float to the top after agitation has ceased.
2. If agitation is not taking place at the time oil is added, the water has a tendency to leach out the emulsifying agent from the oil and prevent it from pulling the oil molecules into the water.

In view of the varied degree of water hardness which can prevail according to source, water conditioning is usually a wise precaution. Very hard water can seriously interfere with emulsification. Water conditioning is preferable prior to addition of lubricant which has nonfoaming and rust inhibitors built in by the manufacturer.

The future of the soluble cutting lubricants is encouraging. It is brightened by the fact that very often such a product can do double duty as an actual machine lubricant. Reduction in fire hazard also is a factor which is decidedly favorable to the use of this type of metal cutting fluid. (*Digested by A. F. Brewer*)

SECTION NEWS

Baltimore—At May 22 annual business meeting, officers for 1959-60 season were installed. They include: C. J. Devries, chairman; H. A. Rau, Jr., vice chairman; H. D. Plumly, secretary, and R. B. McNatt, treasurer. . . . **Detroit**—A conference on "Planned Lubrication as a Profitable Investment" was sponsored by the Detroit section of ASLE, ASME, Wayne State University and the University of Michigan on April 29. The conference was held mainly for industrial executives responsible for control of production costs and quality. . . . **Houston**—J. D. Lykins discussed the activities of ASLE and followed up with a slide presentation on "Plant Lubrication" for 37 members and guests at April 6 meeting. . . . **Hudson-Mohawk**—Considerations in bearing selection, properties, materials and their limitations were covered in Ralph E. Lee's talk "Selection and Application of Bearing Materials" April 25. . . . **Los Angeles**—Those present at April 15 meeting heard Shell Development Co.'s J. D. Accinelli discuss "High-Speed, High-Temperature Ball Bearings." . . . **Milwaukee**—Guest speakers at April 16 meeting were E. J. Kub, and D. Edgcomb of Babcock & Wilcox Co. New officers were elected. **New York**—Golf outing is planned for June 26. Members at May 13 meeting heard panel discussion "Lubrication of Plain Bearings" with speakers B. M. Dunham, Sun Oil Co., N. A. Bracht, Alan Wood Steel Co., and S. S. Connor, Bound Brook Oil-Less Bearing Co. . . . **Northern California**—A one day course on Practical Lubrication designed to give a basic knowledge of lubrication and a respect for what correct lubrication can accomplish, was held on May 19. . . . **Toronto**—Officers elected at April 17 meeting include J. G. Holland, vice-chairman; A. G. Almack, secretary; R. W. White, treasurer, and C. H. Crowe and E. V. Briceland, directors. . . . **Twin Cities**—K. D. Relyea, Texas Research Center, spoke on "Railway Diesel Lubrication" on April 30.

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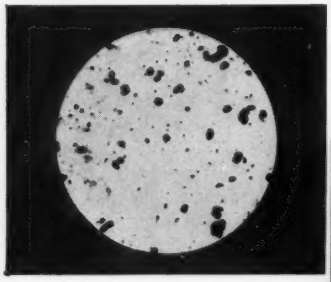
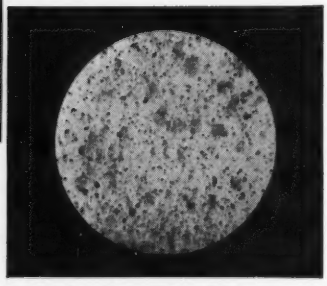


Shown here is a dramatic comparison of particle sizes. Both Acheson's new dispersion, 'dag' 208, (below) and the finest commercially available molybdenum disulfide powder (at left), have been magnified 300 times.



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LUBRICATION ABSTRACTS

(*Cont. from p. 256*)

"Evaluation and Performance of Turbine Oils," by G. H. von Fuchs (Niagara Falls, N. Y.) A.S.T.M. Special Technical Publication, n 211, 1957, pp 93-106.

Rust inhibition properties of alkylated succinic acids are detailed, and the beneficial effect of moisture on them is described. Oxidation stability is reduced by these additives, due to formation of oil-soluble copper and iron soaps. Formation of rust and sludge in the A.S.T.M. T.O.S.T. is shown to correlate with corrosiveness towards copper in the Frigidaire Test. Better performance of the commonly used anti-oxidant 2,6-ditertiary butyl 4-methylphenol in recent years in regard to rusting is attributed to elimination of a monobutylated contaminant. (*Abstracted by M. I. Smith*)

"Filtration of Diesel Engine Oils," J. R. McCoy, (Indiana Commercial Filters, Lebanon, Ind.) ASME paper, n 58-OGP-4 for meeting May 11-15, 1958.

The main function of an oil filter is to reduce wear. Its secondary purpose is to extend the period between oil changes. Severity of operation, type of fuel, and type of oil influence the kind of filter used and its method of installation. Contaminants such as dust, soot, metalic particles, acid, etc. may come from outside or may be formed in the engine. The size of filters depends on many factors, but mainly on horsepower and quality of fuel. Large stationary, low-speed engines can use straight mineral oils and a fullers-earth type filter can be installed. Mobile high output engines require additive type oils and an inert filter is needed since the fullers-earth may adsorb some of the additive. Drawings illustrate four methods of filter installations: (1) by-pass or bleeder type; (2) shunt; (3) full-flow; and (4) utilization of an auxiliary pump and motor. The filter should be placed after the pressure pump to insure full flow, and before the cooler because hot oil filters better than cold. Sometimes the oil is removed and reclaimed since oil does not wear out but is made useless due to contamination. (*Abstracted by A. S. Gressel*)

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Free Fatty Acid (as Oleic)	10.0 - 14.0%	Iodine Value (Hanus)	20.0 - 35.0
Moisture	0.5 - 1.2%	Saponification Number	100.0 - 116.0
Ash	0.1% max.	Penetration at 77°F. unworked	150 - 180 mm/10
Melting Point	36.0° - 44.0°C.	worked	340 - 370 mm/10
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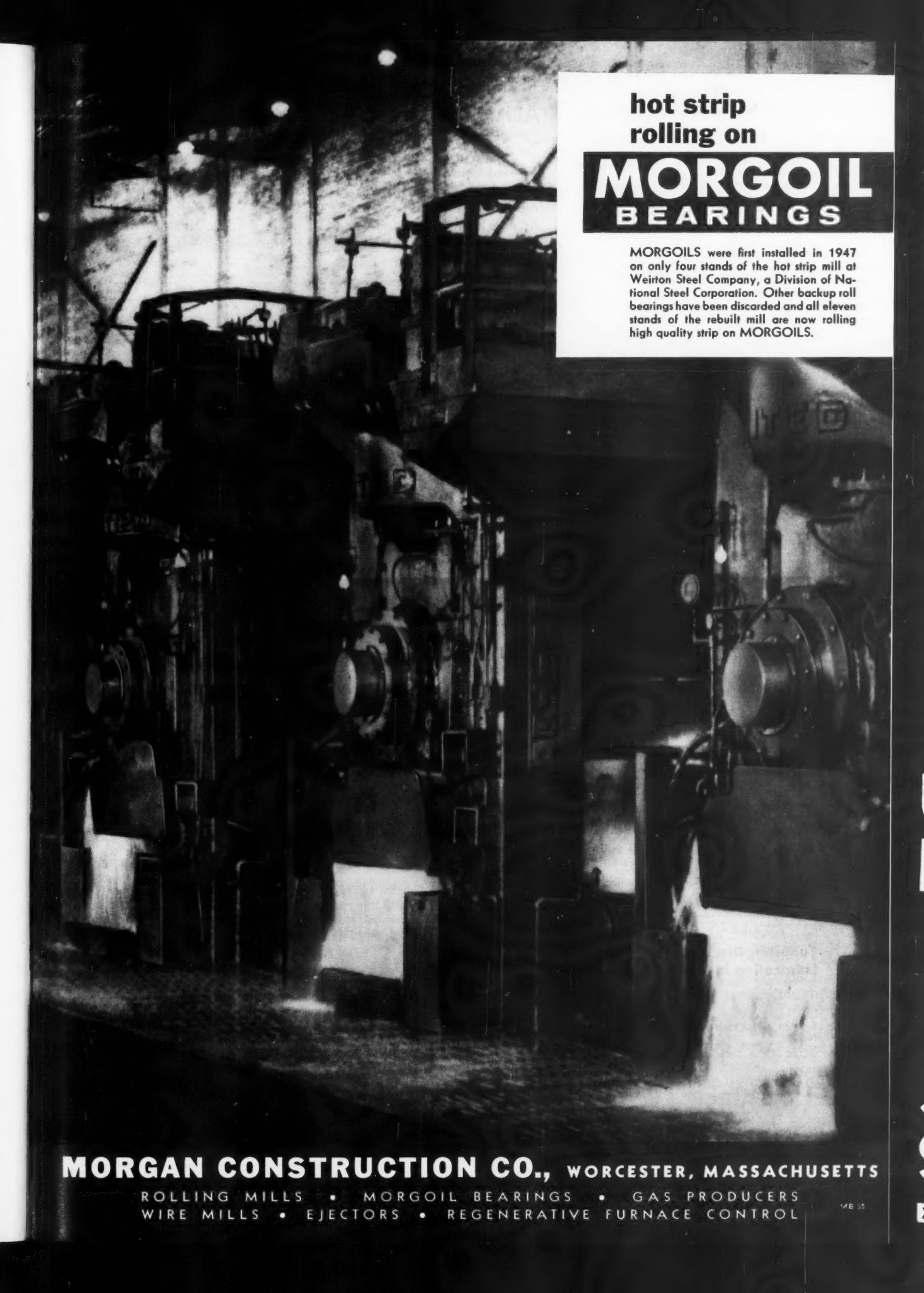
"Factors Causing Wear of Oil Lubricated Brass Bearings," Kichire Endo. (Department of Engr., Hiroshima University) v 1, n 1, Memoirs of the Faculty of Engineering, Hiroshima U., Japan, Dec., 1957.

The wear of 12 different bronzes (1 cm^2 bearing area) on steel at 30 kg/cm^2 and at 75 m/min. is studied. When the acid no. of the lubricating oil is 1.5 wear is 2.7 mg as compared with 1.0 mg for an oil of 0.05 acid no. The reaction rate constant k, for the oxidation of the mineral oil is related to S (fatigue strength of the bronze, proportional to hardness) and S', (the corrosion fatigue of the bronze) by $S' \alpha \frac{S}{K}$. The increases in acid No. caused by adding 5 gr. of the powdered bronzes as catalysts to 40 ml. of the lubricating oil and heating for 2 hr at 180 range from 0.027 for no catalyst to 0.243 for phosphor bronze. Increases in corrosive wear from heating the lubricant to 100C are about 15 times those for runs at room temperature. The relations between unit bearing pressures and temperatures of the brass specimens 1 mm below contact surfaces, or with wear rates, are not linear. (*Abstracted by E. N. Klemgard*)

"Coolant Filtration Improves Product Quality," by V. G. Drake (Fairfield Mfg. Co., Lafayette, Ind.) Tool Engr. v 40, n 6, 1958, pp 101-2.

A viscous S.A.E. 250 oil was used in a battery of ten gear shavers. Each shaver had to be cleaned at least every 24 hours, causing extra labor and down time of the machines. In addition the fine chips packed into the serrations of the shaving tools. The installation of a filter eliminated all that and also improved the surface finish. Photos and description of the filter are included. (*Abstracted by A. S. Gressel*)



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Designing and Testing Journal Box Seals

(*Cont. from p. 245*)

tically an impossibility under present conditions. If controls were established to limit the movement of the axle relative to the journal box, maintain the diameter and finish of the seal fit section of the axle and reduce the tolerances allowed in the journal box casting, the problem of sealing the journal box would be greatly reduced. The cost of these controls would be high and a complete redesign of the truck would probably be less expensive.

Tungsten Disulfide Lubrication in Vacuum

(*Cont. from p. 249*)

ACKNOWLEDGMENT

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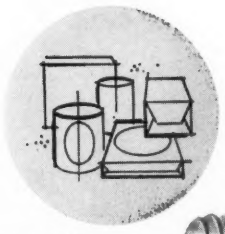

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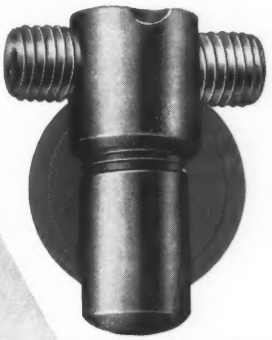

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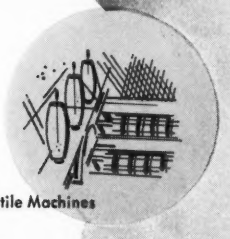

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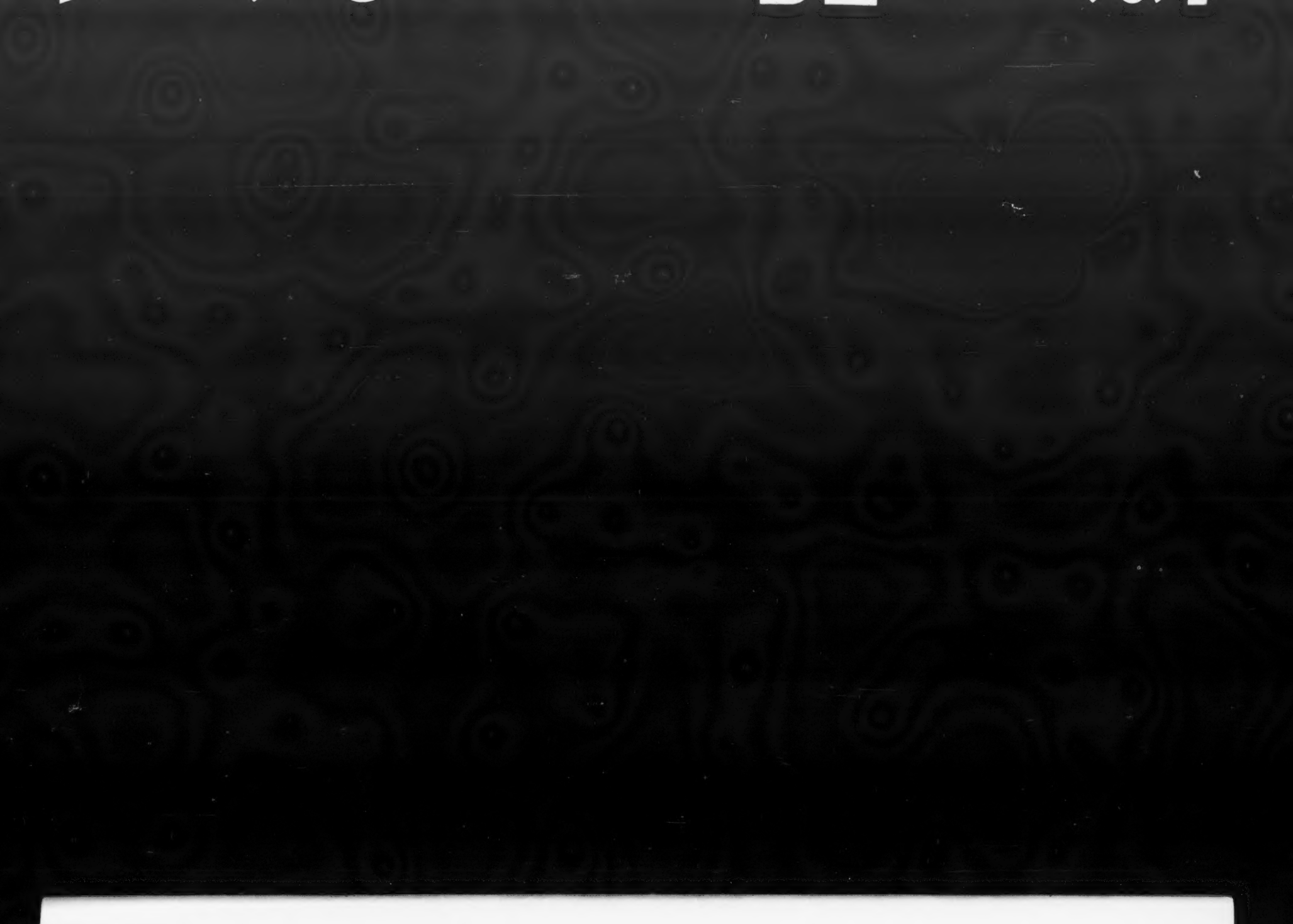
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